Hydro-electric Machines

Hydro-mechanical Equipment

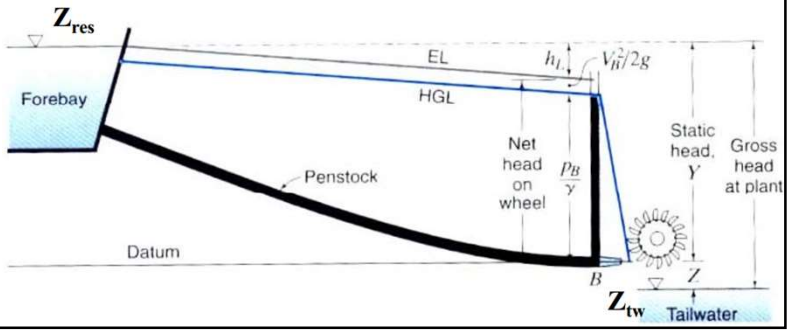
Hydraulic Turbines:

- Mechanical device that converts the potential energy contained in an elevated body of water (a stream or reservoir) into rotational mechanical energy
- Primary function is to drive a electric generator

Types of Hydraulic Turbines:

A) According to the energy conversion

1. Impulse Turbine:



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1. Impulse Turbine:

- The flow energy is completely converted to kinetic energy before transformation in the runner
- The impulse forces being transformed by the direction changes of the flow velocity vectors when passing the buckets create the energy converted to mechanical energy on the turbine shaft
- The flow enters the runner from jets spaced around the rim of the runners; and the jet hits momentarily only a part of the circumference of the runner.
- Pelton turbine and Turgo turbine

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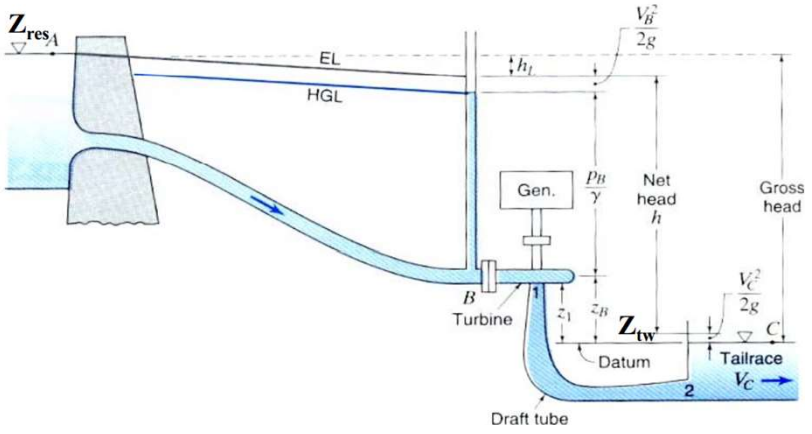
Types of Hydraulic Turbines:

According to the energy conversion

2. Reaction Turbine:



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2. Reaction Turbine:

- Two effects cause the energy transfer from the flow to the mechanical energy on the turbine shaft:
  - Firstly, it follows from a drop in pressure from inlet to outlet of the runner. This is denoted as the reaction part of the energy conversion.
  - Secondly the changes in the directions of the flow velocity vectors through the runner blade channels transfer impulse forces. This is denoted as the impulse part of the energy conversion.
- The pressure drop from inlet to outlet of the runners is obtained because the runners are completely filled with water.
- Francis turbine, Kaplan turbine and Propeller turbine

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Difference between Impulse and Reaction turbine

Aspects	Impulse	Reaction
Conversion of fluid energy	Converted into KE via nozzle	Partly transformed into KE before it enters the runner of turbine
Changes in pressure and energy	The pressure remains atmospheric throughout the action of water in runner	After entering the runner with an excess pressure, water undergoes changes both in velocity and pressure while passing through runner
Water tight case	Not required	Essential
Interaction with water	Wheel does not run full and air has free access to buckets	Water completely fills and all the passages between blades and while flowing between inlet and outlet sections does work on blade
Install of units	Always installed above TWL	Can be installed below TWL
Draft tube	Not used	Necessary to recover lost energy
Flow Regulation	By means of spear valve or deflector fitted into the nozzle	By the adjustment of wicket gates (guide vanes)
Specific speed	Low (7-20)	Moderate to high (50-300:Francis; 240-920: Kaplan)

B) According to Direction of Flow

1. Tangential flow turbine:
   - The flow of water strikes the runner in the direction of tangent to the wheel
   - Example: Pelton wheel turbine

2. Axial flow turbine:
   - The flow of water is in the direction parallel to the axis of the shaft
   - Example: Kaplan turbine and Propeller turbine

3. Radial flow turbine:
   - The flow of water strikes in the radial direction
   - Example: Old Francis turbine

4. Mixed flow turbine:
   - The water enters the runner in the radial direction and leaves in axial direction
   - Example: Modern Francis turbine

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Types of Hydraulic Turbines:

C) According to name of Inventor

1. Pelton wheel turbine
2. Francis turbine
3. Kaplan turbine
4. Deraiz turbine

D) According to Head

1. Very high head:
   > 500 m – Pelton wheel turbine
2. High head:
   71-500 m – Pelton wheel turbine and Francis turbine
3. Medium head:
   16-70 m – Francis turbine and Kaplan turbine
4. Low head:
   2-15 m – Kaplan series

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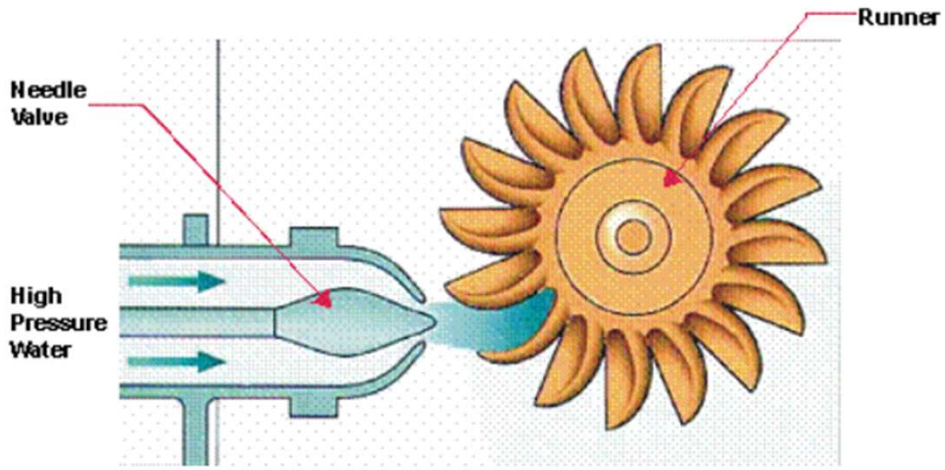
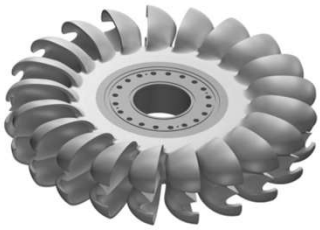
Types of Hydraulic Turbines:

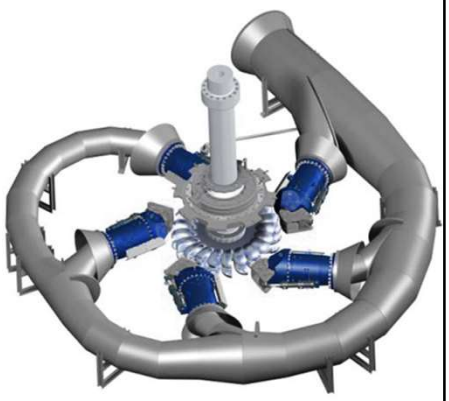
1. Pelton wheel turbine:





Runner

5-jet Pelton turbine



Pelton Turbine

5-jet turbine

- Invented by Lester Allan Pelton in the 1870s
- An impulse type water turbine
- Extracts energy from the impulse of moving water
- The pressure all over the wheel is constant and equal to atmospheric pressure
- The water flows along thetangent to the path of the runner
- Nozzles direct forceful streams of water against a series of spoon-shaped buckets mounted around the edge of a wheel
- As water flows into the bucket, the direction of the water velocity changes to follow the contour of the bucket
- When the water-jet contacts the bucket, the water exerts pressure on the bucket and the water is decelerated as it does a "U-turn“ and flows out the other side of the bucket at low velocity
- Casing is provided to prevent splashing of the water and to discharge water to tailrace
- Suitable for high head plants
- Used in 60 MW Khimti HPP (H=650m), 60 MW Kulekhani I (H=550m) and 32 MW Kulekhani II (H=284.1 m)

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2. Turgo turbine:

- The Turgo turbine is similar to the Pelton but the jet strikes the plane of the runner at an angle (typically 20° to 25°) so that the water enters the runner on one side and exits on the other.
- Therefore the flow rate is not limited by the discharged fluid interfering with the incoming jet (as is the case with Pelton turbines).
- As a consequence, a Turgo turbine can have a smaller diameter runner and rotate faster than a Pelton for an equivalent flow rate.
- The Turgo turbine is an impulse water turbine designed for medium head applications.
- In factory and lab tests Turgo Turbines perform with efficiencies of up to 90%.
- Complex blade design but greater flow possibilities.

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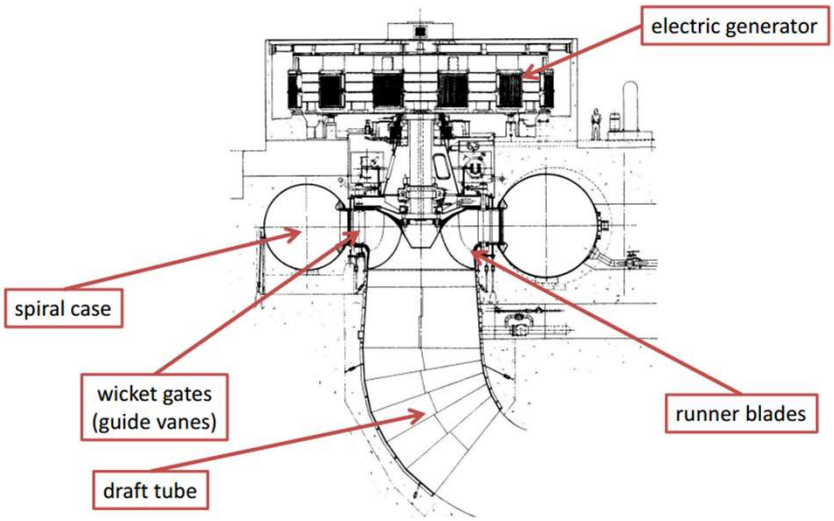
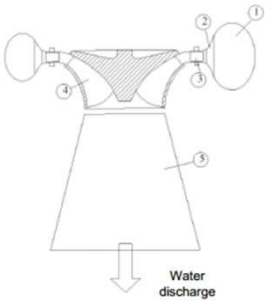
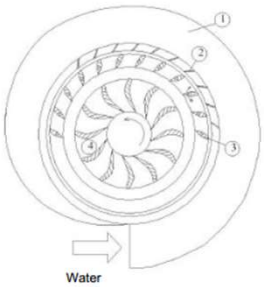
3. Francis turbine:

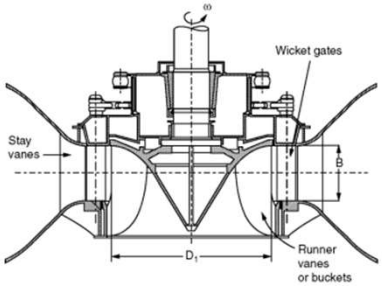


Main components

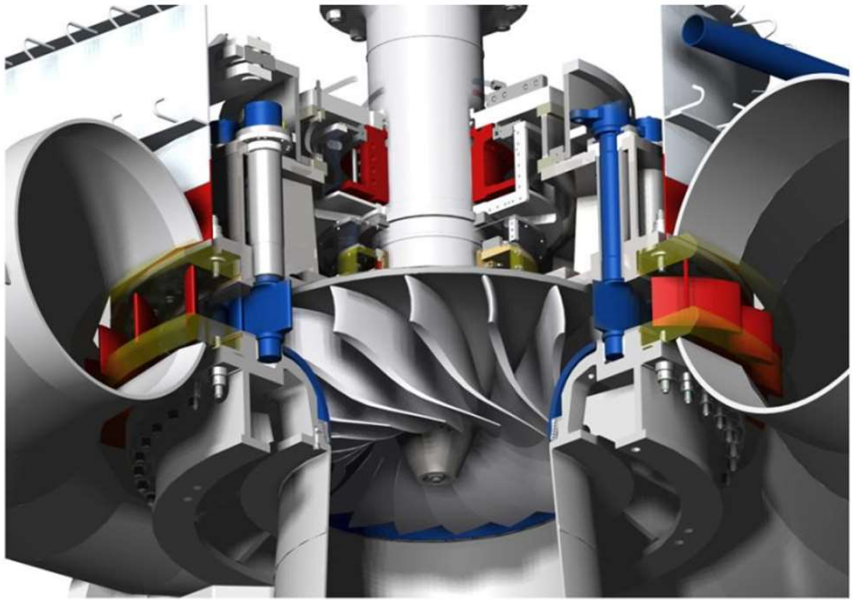




1. spiral case
2. stay vanes
3. wicket gates (guide vanes)
4. runner
5. draft tube



Main components



- Developed by James B. Francis in 1984
- It is an inward-flow reaction turbine that combines radial and axial flow concepts
- Operates at its best completely filled with water at all times
- The most common water turbine in use today
- Operates in a water head from 40 to 600 m
- Power output ranges just a few kilowatts up to 800 MW
- The speed range of the turbine is from 75 to 1000 rpm
- Wicket gates around the outside of the turbine's rotating runner control the rate of water flow through the turbine for different power production rates
- The runner consists of 12-22 nos of curved blades
- Water is discharged to the tailrace through a closed tube of gradually enlarged section, called draft tube
- Used in 144 MW Kali Gandaki HPP (H=115 m), 69 MW Marsyangdi (H=95 m); both have three units of Francis turbine

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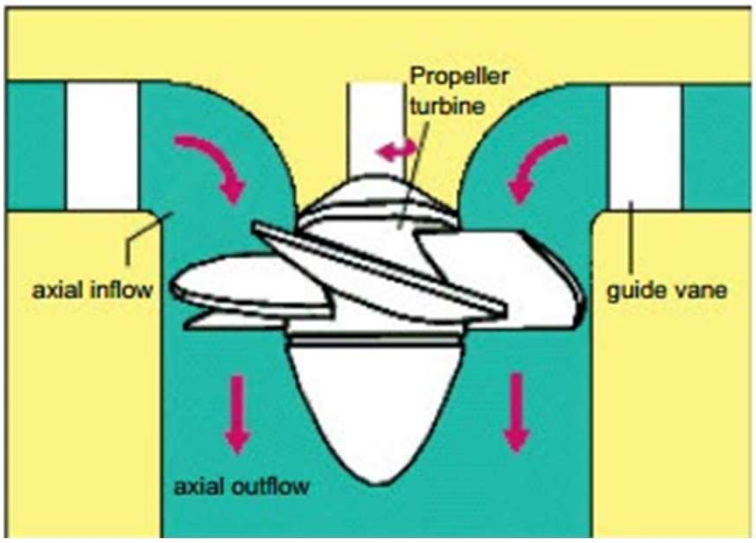
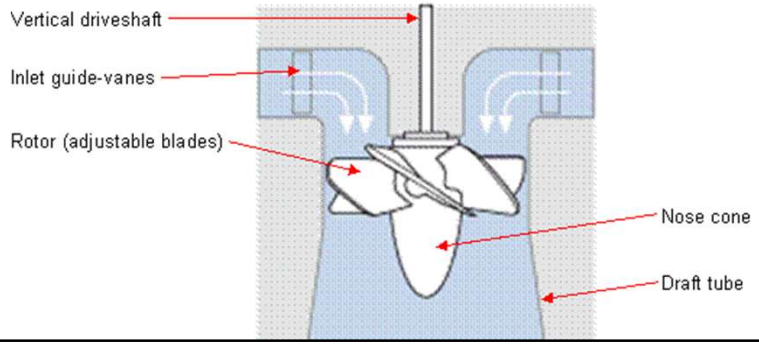
4. Propeller turbine:



➢Propeller turbines have non-adjustable propeller vanes. They are used in where the range of flow / power is not large. Larger propeller turbines produce more than 100 MW
➢Axial flow turbine
➢No of blades: 3-8
➢The head ranges from 4–80 m
➢Runner diameter: 2-11 m

5. Kaplan turbine:



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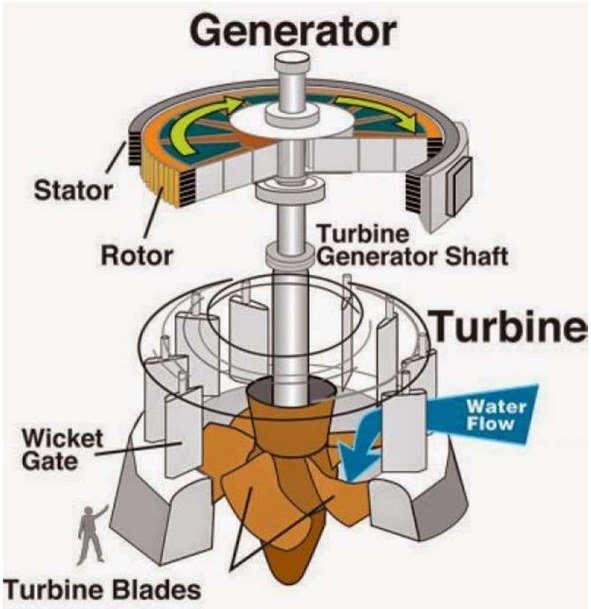

- Axial flow turbine
- No of blades: 3-8
- Kaplan turbine is developed by Viktor Kaplan in 1913
- Kaplan turbine is the propeller-type water turbine which has adjustable blades; combined automatically adjusted propeller blades with automatically adjusted wicket gates to achieve efficiency over a wide range of flow and water level
  - Large range of load (50% under load - 50% over load)
- The Kaplan turbine was an evolution of the Francis turbine; allowed efficient power production in low-head applications that was not possible with Francis turbine
  - The head ranges from 10–70 m and the output from 5 to 200 MW
  - Runner diameter: 2-11 m
- Kaplan turbines are now widely used throughout world in high-flow, low-head power production

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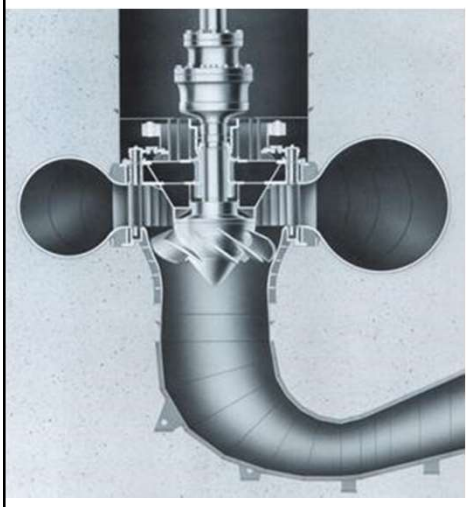
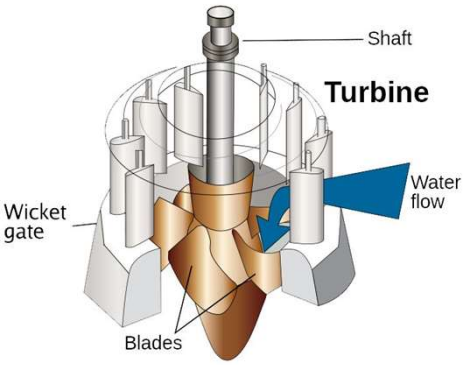
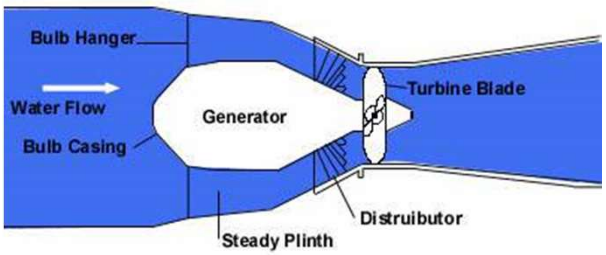
6. Deriaz turbine:





- Invented by Paul Deriaz in 1956
- Mixed-flow reaction turbine
- Similar to a Kaplan turbine but inclined blades (flow over the runner is at 45°) to make it more suitable for higher heads
- Particularly suitable for the head range between 20 - 100 m, in between the ranges of Francis and Kaplan turbines
- No of blades: 10-12

7. Bulb turbine:



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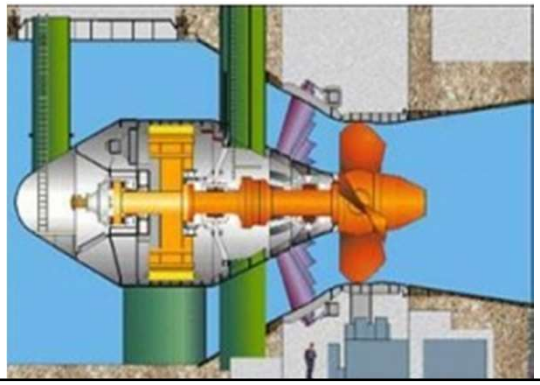
- Generator is housed in enclosed bulb-shaped casing
- Axial flow reaction turbine
- immersed in the water channel, the flow enters and exits the turbine with minor changes in direction
- The draft tube is straight flaring tube
- Compact arrangement of components and less danger of cavitations
- The head ranges from 3–20 m



Turbine Efficiency

Hydraulic Efficiency: The hydraulic efficiency of the turbine is the ratio of the power developed by the runner to the net power supplied by the water at the entrance to the turbine.

$$\eta_h = \frac{\text{power developed by runner}}{\text{net power supplied at the runner}}$$

Mechanical Efficiency: The mechanical efficiency of the turbine is the ratio of the power available at the turbine shaft to the power developed by the runner. These two differ by an amount of the mechanical losses viz. bearing loss.

$$\eta_m = \frac{\text{power available at the turbine shaft}}{\text{power developed by the runner}}$$

Volumetric Efficiency: The volumetric efficiency is the ratio of quantity of water actually striking the runner and the quantity of the water supplied to the turbine. These two quantities differ by an amount of water that slips directly to the tailrace without striking runner.

$$\eta_v = \frac{Q}{Q + \Delta Q}$$

Overall Efficiency: The overall efficiency of the turbine is the ratio of the power available at the turbine shaft to the power supplied at the entrance to the turbine.

$$\eta_o = \eta_h \times \eta_v \times \eta_m$$

Turbine Speeds

1. Runway speed

If the external load on the machine suddenly drops to zero (sudden rejection) and the governing mechanism fails at the same time, the turbine will tend to race up the maximum possible speed, known as runway speed. The suggested runway speed of the various runners for their appropriate design, and acceptable head variations of such turbines are given in following table.

Type of Runner	Runaway speed (% of normal speed)	Acceptable head variation (% of design head)	
		Minimum	Maximum
Impulse (Pelton)	170-190	65	125
Francis	200-220	50	150
Propeller	250-300	50	150

2. Specific speed (Ns)

The specific speed of a turbine is defined as a speed of geometrical similar turbine that would develop unit power under unit head. All geometrical similar turbine (irrespective of size) will have same specific speed when operating under same head.

$$N_S = \frac{P^{1/2} N}{H^{5/4}}$$

where,

N = Rotational speed of turbine in rpm
P = Power out-put
H = Head of turbine

Specific speed (Ns) for different type of turbines

Given below the specific speed of different type of turbines that would develop 1 HP under 1 m head.

$$N_s = \frac{2400}{\sqrt{H}}$$ for Francis Turbine

$$N_s = \frac{1080}{H^{\frac{1}{4}}}$$ for Fixed Blade Propeller Turbine

$$N_s = \frac{1475}{H^{\frac{1}{3}}}$$ for Adjustable Blade Propeller (Kaplan) Turbine

3. Synchronous speed

If the turbine is directly connected to the generator, the turbine speed η must be a synchronous speed. For the turbine speed n to be synchronous, the following equation must be fulfilled.

$$n = \frac{120f}{N_p}$$

Where n= rotational speed

f = electrical frequency in Hertz, Hz (50Hz in Nepal)

N_p = number of generator pole.

(divisible by 4 for head up to 200 m or by 2 for heads above 200 m)

4. Speed factor or speed ratio (ø)

It is the ratio of peripheral speed 'v' of the buckets or vanes at the nominal diameter 'D' to the theoretical velocity of water under the effective head 'H' acting on the turbine.

$Ø = v/\sqrt{(2gH)} = \pi D N/(60\sqrt{(2gH)})$

N= Angular velocity in rpm

Range of ϕ values, specific speeds and heads

Type of Runner	ϕ	N_s	H(m)	Efficiency (%)
Impulse	0.43-0.48	8-17 17 17-30	>250	85-90 90 90-82
Francis	0.60-0.90	40-130 130-350 350-452	25-450	90-94 94 94-93
Propeller	1.4-2.0	380-600 600-902	<60	94 94-85

18·10. Unit Quantities

Let us consider a *single unit*. When the head on the unit is changed/varied then the speed of an ungoverned turbine changes. The velocities at various points do not change direction but their magnitudes vary in proportion to the *square root of the head.*

At a given point in the turbine under a head H, let

V = Absolute velocity,

V_r = Relative velocity,

u = Peripheral velocity, and

V', V_r', u' = Corresponding values at a different head H', then as velocity is proportional to $\sqrt{H}$, we have

$$\frac{u}{u'} = \frac{V_r}{V_r'} = \frac{V}{V'} = \frac{\sqrt{H}}{\sqrt{H'}} \quad \text{...(18·34)}$$

If the discharges are Q and Q' then,

$$\frac{Q}{Q'} = \frac{V}{V'} = \frac{N}{N'} = \frac{\sqrt{H}}{\sqrt{H'}} \quad \text{...(18·35)}$$

If the power outputs are P and P' then,

$$\frac{P}{P'} = \frac{QH}{Q'H'} = \frac{\sqrt{H}}{\sqrt{H'}} \times \frac{H}{H'} = \left(\frac{H}{H'}\right)^{3/2} \quad \text{...(18·36)}$$

$$\left(\because \frac{Q}{Q'} = \frac{\sqrt{H}}{\sqrt{H'}}\right)$$

The hydraulic efficiency of the turbine under these two heads may be considered to be nearly same, as the velocity triangles at these heads are similar at a point.

If the various quantities are *reduced to a theoretical one metre head* the comparison of performance data and computations of experimental values on a single unit are *considerably simplified*.

Then,
$$N_u = \frac{N}{\sqrt{H}} \quad ...(18\cdot37)$$

$$Q_u = \frac{Q}{\sqrt{H}} \quad ...(18\cdot38)$$

$$P_u = \frac{P}{H^{3/2}} \quad ...(18\cdot39)$$

The above quantities are called **unit quantities** of a turbine. *Unit speed is the hypothetical speed of the turbine operating under one metre head*. Similarly, other proportionality constants in Eqns. 18·38 and 18·39 are defined.

For presenting the performance of geometrically similar turbines independent of the actual head, discharge and power output the ***unit characteristics*** prove quite helpful. **Geometrically similar turbines will have the same unit characteristics under similar operating conditions**. Thus with the help of a model the performance of a prototype can be predicted within certain limits.

If a turbine is working under different heads the behaviour of the turbine can be easily known from the values of the *unit quantities* as follows :

Let, H_1, H_2 = Heads under which a turbine works,
N_1, N_2 = Corresponding speeds,
Q_1, Q_2 = Corresponding discharges, and
P_1, P_2 = Corresponding powers developed.

Then using eqns. (18·37), (18·38), (18·39), respectively, we obtain

$$N_u = \frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}} \quad ...(18\cdot40)$$

$$Q_u = \frac{Q_1}{\sqrt{H_1}} = \frac{Q_2}{\sqrt{H_2}} \quad ...(18\cdot41)$$

$$P_u = \frac{P_1}{H_1^{3/2}} = \frac{P_2}{H_2^{3/2}} \quad ...(18\cdot42)$$

Example 18·49. *A turbine is to operate under a head of 25 m at 200 r.p.m. The discharge is 9 m³/s. If the efficiency is 90 per cent determine the performance of turbine under a head of 20 m.* [M.U]

Solution. Head under which turbine works, $H_1 = 25$ m

Speed of the turbine, $N_1 = 200$ r.p.m.

Discharge through the turbine, $Q_1 = 9$ m³/s

Efficiency (overall), $\eta_0 = 90\%$

Performance of turbine under a head of 20 m; N_2, Q_2, P_2 :

Performance of turbine under a head, $H_2 = 20$ m means to find speed (N_2), discharge (Q_2), and power generated (P_2) by the turbine when working under a head of 20 m.

Overall efficiency, $\eta_0 = \frac{\text{Shaft power}}{\text{Water power}} = \frac{P}{wQH} = \frac{P_1}{wQ_1H_1}$

$\therefore$ $P_1 = \eta_0 \times wQ_1H_1 = 0{\cdot}9 \times 9{\cdot}81 \times 9 \times 25 = 1986{\cdot}5$ kW

Now, $\frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}}$ ...[Eqn. (18·40)]

$\therefore \quad N_2 = \frac{N_1\sqrt{H_2}}{\sqrt{H_1}} = \frac{200 \times \sqrt{20}}{\sqrt{25}} = 178{\cdot}88$ r.p.m. **(Ans.)**

and, $\frac{Q_1}{\sqrt{H_1}} = \frac{Q_2}{\sqrt{H_2}}$ ...[Eqn. (18·41)]

$\therefore \quad Q_2 = \frac{Q_1\sqrt{H_2}}{\sqrt{H_1}} = \frac{9 \times \sqrt{20}}{\sqrt{25}} = 8{\cdot}05\ m^3/s$ **(Ans.)**

and, $\frac{P_1}{H_1^{3/2}} = \frac{P_2}{(H_2)^{3/2}}$ ...[Eqn (18·42)]

$\therefore \quad P_2 = \frac{P_1 \times (H_2)^{3/2}}{(H_1)^{3/2}} = \frac{1986{\cdot}5 \times (20)^{3/2}}{(25)^{3/2}} = 1421{\cdot}4$ kW **(Ans.)**

Performance of Turbines under unit quantities

The unit quantities give the speed, discharge and power for a particular turbine under a head of 1m assuming the same efficiency. Unit quantities are used to predict the performance of turbine.

1. Unit speed (N_u) - Speed of the turbine, working under unit head
2. Unit power (P_u) - Power developed by a turbine, working under a unit head
3. Unit discharge (Q_u) - The discharge of the turbine working under a unit head

18·13. Performance Characteristics of Hydraulic Turbines

The turbines are normally designed for specific values of head, speed, discharge, power and efficiency (known as the *designed conditions*). But oftenly turbines may be required to operate under conditions different from those for which these have been designed. Thus, to know about their exact behaviour under varying conditions it becomes necessary to conduct tests either on the actual turbines at the site or on their small scale models in a research laboratory. The results so obtained are usually represented graphically and the curves obtained are known as "*Characteristic curves*". These curves are usually plotted in terms of unit quantities (for sake of convenience). The characteristic curves are of the following types :

1. Main or constant head characteristic curves.
2. Operating or constant speed characteristic curves.
3. Constant efficiency or iso-efficiency or Muschel curves.

18·13·1. Main or Constant Head Characteristic Curves

- *Head and gate opening are maintained constant.*
- *Speed is varied* by allowing a variable quantity of water to flow through the inlet opening.
- The brake power (P) is then measured mechanically by means of a dynamometer.
- The overall efficiency and unit quantities are then calculated by using the basic data; these are then plotted *against unit speed as abscissa.*

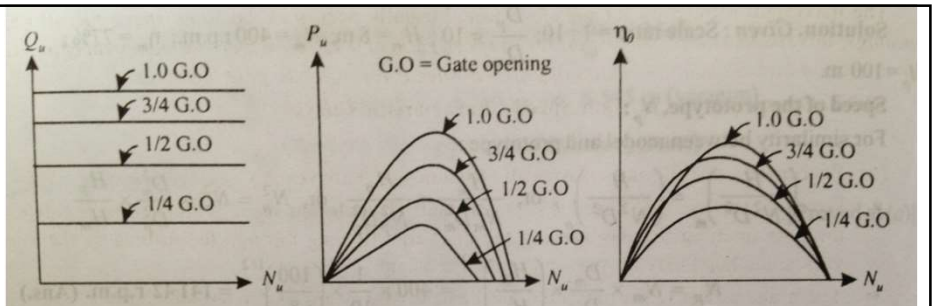


Fig. 18.53. Main characteristic curves of Pelton wheel.

Figs. 18·53, 18·54 and 18·55 show the main characteristic curves of Pelton wheel, Francis turbine and Kaplan turbine respectively.

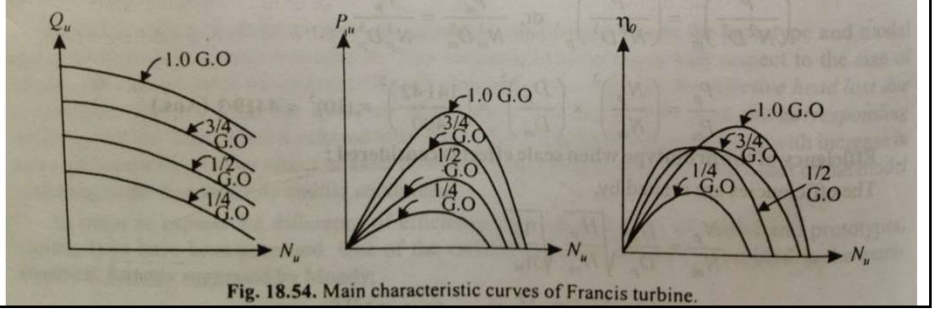


Fig. 18.54. Main characteristic curves of Francis turbine.

The main characteristic curves yield the following information :

- The discharge Q_u for a *Pelton wheel* depends only upon the gate opening and is independent of N_u; the curves for Q_u are horizontal.
- The curves between Q_u and N_u for a *Francis turbine are falling curves*. This is due to the fact that a *centrifugal head develops which* acts outwards and *opposes* the external head causing flow, eventually decreasing the discharge as the speed increases.
- The curves between Q_u and N_u for a *Kaplan turbine* are *rising curves*; the discharge increases with the increase in speed.

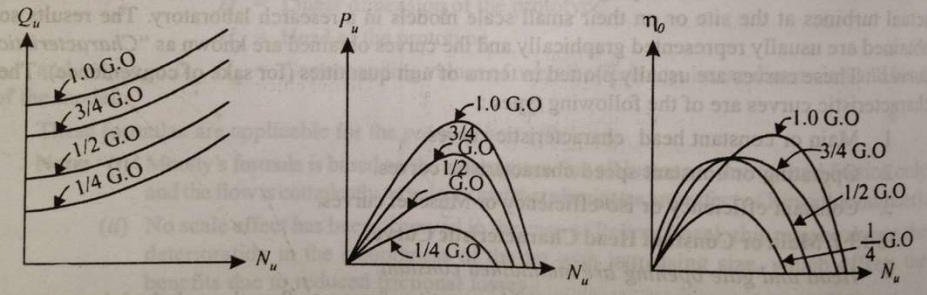


Fig. 18.55. Main characteristic curves of Kaplan turbine.

- The curves between P_u and N_u and those between η_0 and N_u indicate that at a particular speed the efficiency is maximum.

The maximum efficiency for a *Pelton wheel* usually occurs at the *same speed for all gate openings; this speed usually corresponds to a speed ratio of 0·45 . However, the maximum efficiency for a reaction turbine usually occurs at different speeds for different gate openings.*

18·3·2. Operating or Constant Speed Characteristic Curves

These curves are obtained as follows :

(*a*) **Percentage of full load *v/s* overall efficiency (η_0) curves :**

- For each gate opening speed is kept constant. The constant speed is attained by regulating the gate opening thereby varying the discharge flowing through the turbine as the load varies; the head may or may not remain constant.
- The brake power (P) is measured mechanically by means of a dynamometer.
- The overall efficiency (η_0) is then calculated from the measured values of discharge, head and power.
- Further knowing the total load capacity of the turbine the percentage of full load is computed from the measured power and a plot of η_0 *v/s* percentage of full load is prepared.

Fig. 18·56 shows the graphs plotted between *percentage of full load v/s* η_0 for different types of turbines. The following *points are worth noting* :

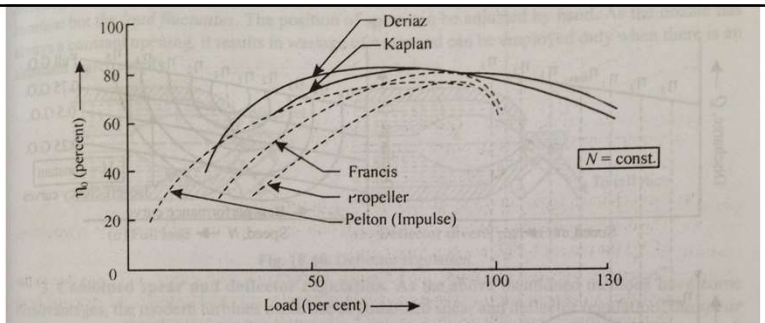


Fig. 18.56. Percentage of full load v/s η_0 curves for hydraulic turbines.

- As the percentage full load increases η_0 also increases (In other words, at reduced loads η_0 is also less).
- At 100 per cent full load η_0 is near about the maximum efficiency in all cases.
- The Kaplan, the Deriaz and the Pelton wheel maintain a high efficiency over a longer range of *part load* as compared with either the Francis or the fixed blade propeller turbine.
- The maximum overall efficiency of all the turbines is almost the same (about 85%).

(*b*) **Overall efficiency (η_0) and output (shaft) power (P) v/s discharge (Q) curves:**

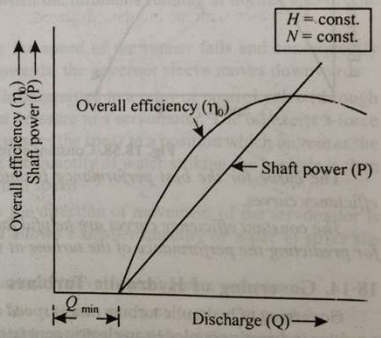


Fig. 18.57. η_0 and P v/s discharge curves.

Fig. 18·57 shows overall efficiency (η_0) and shaft power (P) *v/s* discharge curves. Q_{min} is the minimum discharge required to set the turbine runner into motion from its state of rest. These curves yield the following information :

- Shaft power or output power (P) is a straight line, since $P \propto Q$ if H (head) is constant.
- η_0 *v/s* discharge (Q) graph is curvilinear and η_0 increases with Q and remains *nearly* constant beyond a particular value of discharge.

18·13·3. Constant efficiency or iso-efficiency or Muschel curves

Refer to Fig. 18·58. As η–N curve is of parabolic nature, there exits two speeds for one value of efficiency except for maximum efficiency which occurs at one speed only. Corresponding to these values of speeds there are also two values of discharge for each value of efficiency (Q-N curve). Hence on Q-N curve we can plot two points for each value of efficiency and one point for maximum efficiency. By adopting this procedure for different gate openings or heads we can get number of Q-N curves and we can plot on them efficiency points (as described above). The points denoting the same efficiency can now be joined to get constant iso-efficiency curves or Muschel curves (The German word 'Muschel' means shell, indicating shape of curve). The diagram showing these curves is also called Hill diagram (since it looks like top view of a hill). In actual practice unit speed and unit discharge are taken along the co-ordinate axes.

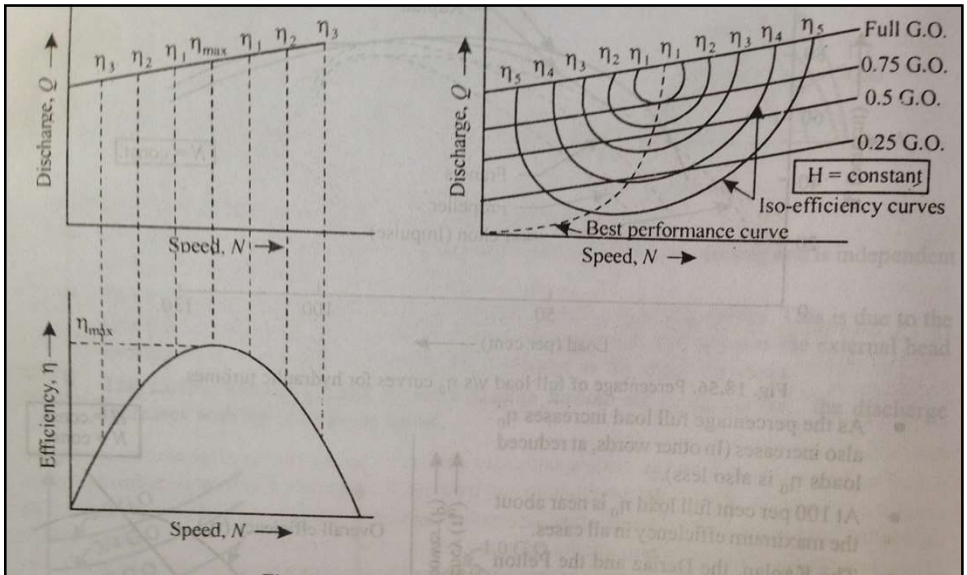
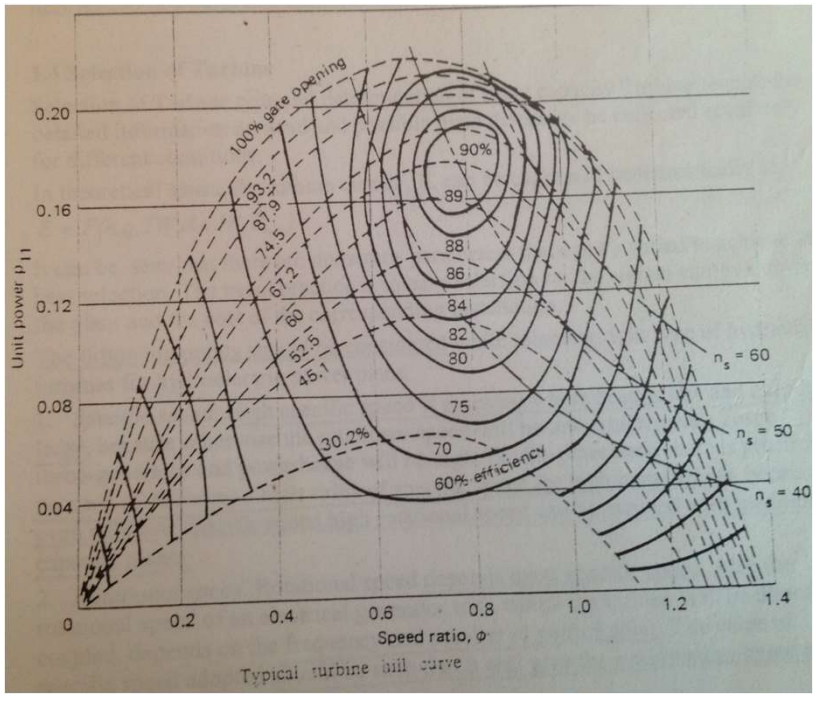
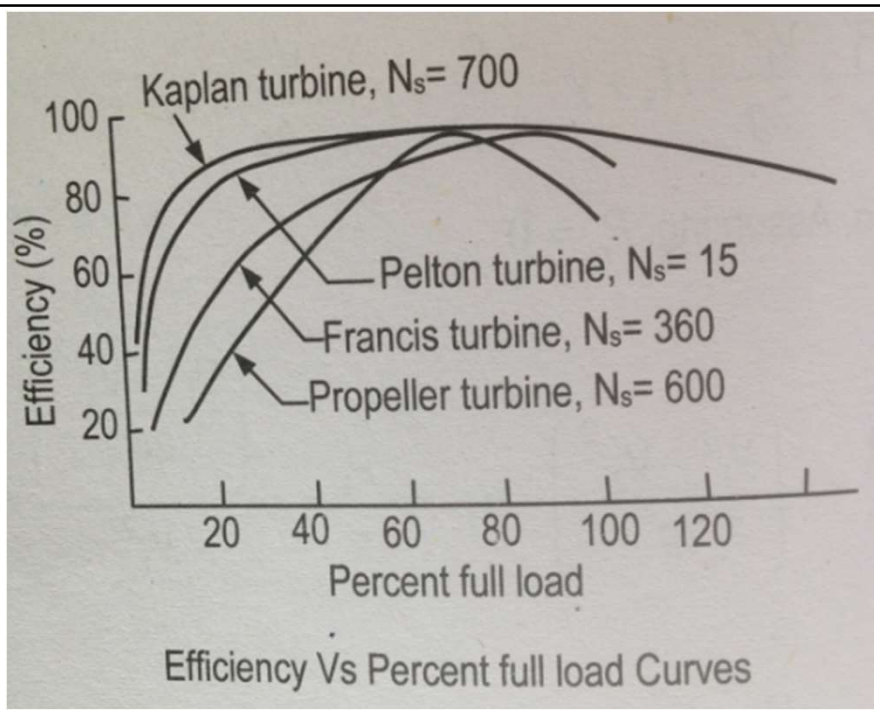


Fig. 18.58. Constant efficiency curves for turbines.

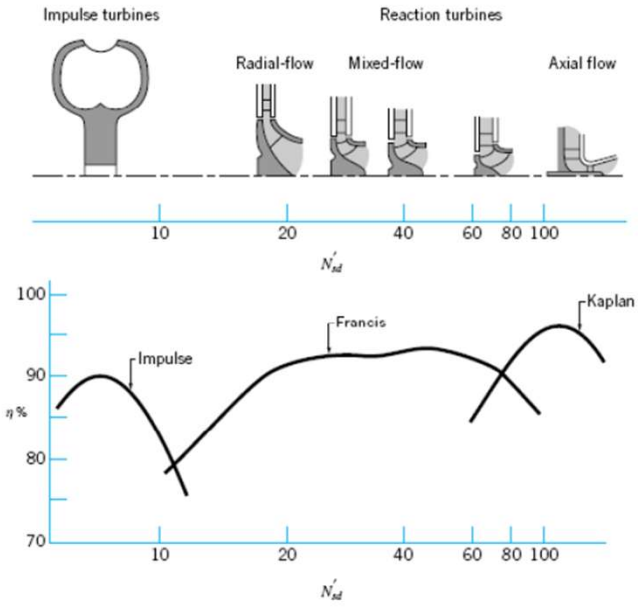
The *curve for the best performance is obtained by joining the peak points of the various efficiency curves.*

The constant efficiency curves are helpful for determining the zone of constant efficiency and for predicting the performance of the turbine at various efficiencies.





Efficiency vs Specific Speed



Draft Tube

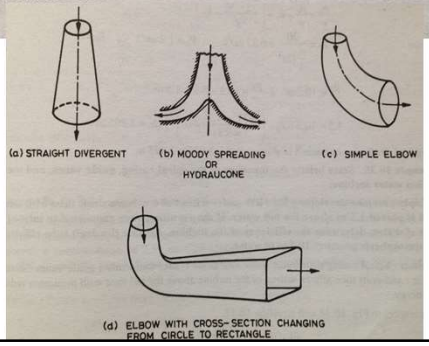
Draft tube is an integeral part of the low head turbines with large through flow, *i.e.*, for reaction turbines of the Francis and Kaplan type. The unit consists an airtight diverging conduit with cross-sectional area increasing along its length. One end of this diverging tube is connected to the runner exit and the other is located below the level of tail race.

A draft tube has the following functions to perform :

- decrease the pressure at the runner exit to a value less than atmospheric pressure and thereby increase the working head.
- recover a portion of the exit kinetic energy which otherwise goes waste to the tail race.

These two factors help the turbine to develop more power and thereby the efficiency of the turbine increases. The unit also serves to fix the turbine above the tail race and that ensures proper inspection of the turbine.

Types of Draft Tube

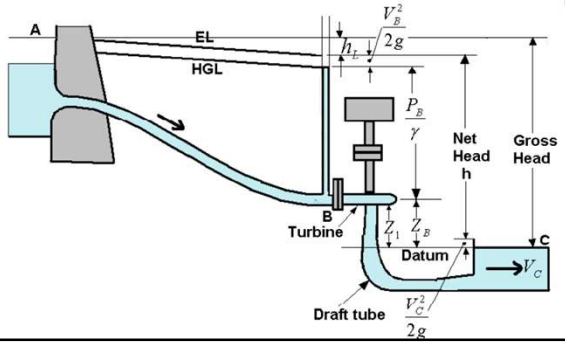

(a) STRAIGHT DIVERGENT
(b) MOODY SPREADING OR HYDRAUCONE
(c) SIMPLE ELBOW
(d) ELBOW WITH CROSS-SECTION CHANGING FROM CIRCLE TO RECTANGLE

* *Straight conical type* which takes the form of the frustrum of a cone with central angle less than 8-degree ($\alpha = 4^{o}$) so as to prevent the possibility of flow separation. This draft tube has an efficiency of about 90% and is generally employed for low specific speed and vertical shaft turbines.

* *Simple elbow type* finds application where the length of the shaft has to be minimum so as to cut short the volume of excavation. Because of bend there is a loss of head and so the elbow type draft tube has a low efficiency of the order of 60%.

* *Elbow type* having a circular cross-section at the inlet and a square or rectangular section at the outlet. The change from circular section in the vertical leg to rectangular section in the horizontal leg takes place in the bend. The draft tube efficiency is about 85%.

* *Moody's spreading draft tube* is provided with a solid central core of conical shape which reduces whirling action of discharge water. The draft tube has an efficiency of about 85% and is suited particularly for helical flows which occur when the water leaves the runner with a whirl component.



Energy Equation Applied to Draft Tube

$$\frac{P_B}{\gamma} + Z_B + \frac{V_B^2}{2g} + h_L$$

- The velocity V_2 can be reduced by having a diverging passage.
- To prevent cavitation, the vertical distance z_1 from the tail water to the draft tube inlet should be limited so that at no point within the draft tube or turbine will the absolute pressure drop to the vapour pressure of water.

Absolute velocity at inlet of draft tube for Francis turbine (m/s): $V = 8.74 + \frac{248}{N_s}$

Absolute velocity at inlet of draft tube for Kaplan turbine (m/s): $V = 8.42 + \frac{250}{N_s}$

Draft-tube theory

Consider a conical draft tube as shown in figure.

Let H_s - Vertical height of draft tube above the tail race.

y - Distance of bottom of draft tube from tail race.

Applying Bernoulli's equation to section 1-1 and 2-2

$$(H_s + y) + \frac{P_1}{\gamma} + \frac{V_1^2}{2g} = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + h_f$$

h_f loss of energy between 1-1 & 2-2

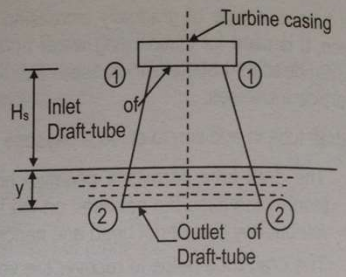


Figure: Draft tube theory

here $\frac{P_2}{\gamma} = \frac{P_a}{\gamma} + y$

or $(H_s + y) + \frac{P_1}{\gamma} + \frac{V_1^2}{2g} = \frac{P_a}{\gamma} + y + \frac{V_2^2}{2g} + h_f$

or $\frac{P_1}{\gamma} = \frac{P_a}{\gamma} + \frac{V_2^2}{2g} + h_f - \frac{V_1^2}{2g} - H_s$

$$\frac{P_1}{\gamma} = \frac{P_a}{\gamma} - H_s - \left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f\right)$$

Hence in this equation $\frac{P_1}{\gamma}$ is less than atmospheric pressure. There is the limiting value for the setting of the runner with respect tail water due to cavitations effect.

Efficiency of draft tube

$$\eta_d = \frac{\text{Atual conversion of kinetic head into pressure head}}{\text{Kinetic head at the inlet of draft tube}}$$

Theoretical conversion of kinetic head into pressure head in draft tube = $(\frac{V_1^2}{2g} - \frac{V_2^2}{2g})$

Actual Conversion of kinetic head into pressure head = $(\frac{V_1^2}{2g} - \frac{V_2^2}{2g}) - h_f$

Draft tube efficiency, $\eta_d = \frac{(\frac{V_1^2}{2g} - \frac{V_2^2}{2g}) - h_f}{V_1^2/2g}$

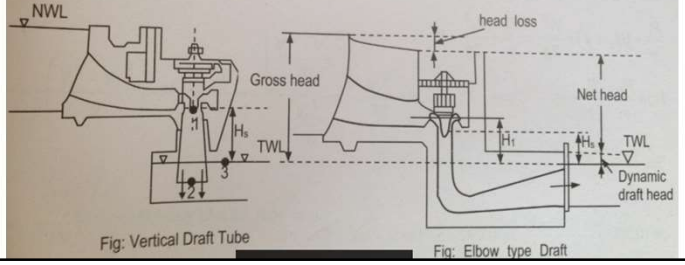


Fig: Vertical Draft Tube

Fig: Elbow type Draft

Turbine scroll case

A scroll case is the conduit directing the water from the intake or penstock to the runner in reaction-type turbine installations (in the case of impulse wheels a casing is usually provided only to prevent splashing of water and to lead water to the tail race). A spiral-shaped scroll case of the correct geometry ensures even distribution of water around the periphery of the runner with the minimum possible eddy formations. The shape and internal dimensions are closely related to the design of the turbine.

(a) Full spiral case

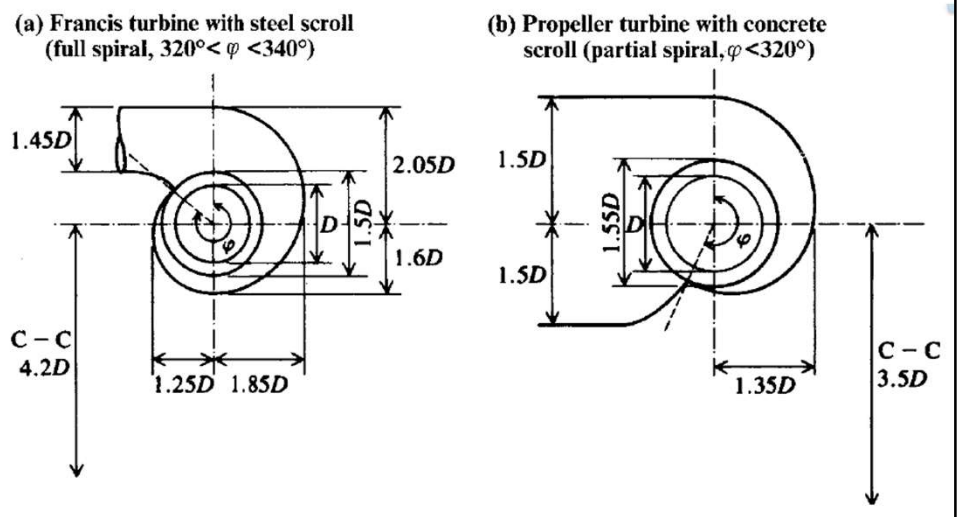
A full spiral case (Figure (a)) entirely enclosing the turbine with a nose angle, ø, of 360° ensures most perfect flow conditions. However, in practice spiral cases with 320° < ø <340° are also called full spiral cases. This kind of spiral case will generally be used in medium and high-head installations where discharge requirements are smaller.

Turbine scroll case

(b) Partial spiral case

For low-head plants the entrance area should be large so as to allow large flows. This is achieved by choosing nose angles that are less than 320°. The spacing of the units is therefore governed by these large entry widths of the partial spiral scroll cases.

For high-head plants, a circular scroll case cross-section is normally adopted; a metal casing is more suited to this shape. For low-head plants where the water quantities are large a rectangular section with rounded corners may be constructed *in situ* in concrete. The approximate dimensions of a Francis turbine layout with steel scroll case and a propeller turbine layout with concrete scroll, respectively, are shown in Figures (a) and (b). The design of the shape of the spiral case is governed by the flow requirements. Initial investigations should be based on the following assumptions: (a) a spiral case of constant height; (b) an evenly distributed flow into the turbine; (c) no friction losses.



Recommended dimensions of scroll casings

Cavitation in Hydraulic Turbine

Cavitation is formation of vapor bubbles in the liquid flowing through any Hydraulic Turbine. Cavitation occurs when the static pressure of the liquid falls below its vapor pressure. Cavitation is most likely to occur near the fast moving blades of the turbines and in the exit region of the turbines.

Causes of Cavitation

The liquid enters hydraulic turbines at high pressure; this pressure is a combination of static and dynamic components. Dynamic pressure of the liquid is by the virtue of flow velocity and the other component, static pressure, is the actual fluid pressure which the fluid applies and which is acted upon it. Static pressure governs the process of vapor bubble formation or boiling. Thus, Cavitation can occur near the fast moving blades of the turbine where local dynamic head increases due to action of blades which causes static pressure to fall. Cavitation also occurs at the exit of the turbine as the liquid has lost major part of its pressure heads and any increase in dynamic head will lead to fall in static pressure causing cavitation.

Detrimental Effects of Cavitation

The formation of vapor bubbles in cavitation is not a major problem in itself but the collapse of these bubbles generates pressure waves, which can be of very high frequencies, causing damage to the machinery. The bubbles collapsing near the machine surface are more damaging and cause erosion on the surfaces called as cavitation erosion. The collapses of smaller bubbles create higher frequency waves than larger bubbles. So, smaller bubbles are more detrimental to the hydraulic machines.

Smaller bubbles may be more detrimental to the hydraulic machine body but they do not cause any significant reduction in the efficiency of the machine. With further decrease in static pressure more number of bubbles is formed and their size also increases. These bubbles coalesce with each other to form larger bubbles and eventually pockets of vapor. This disturbs the liquid flow and causes flow separation which reduces the machine performance sharply. Cavitation is an important factor to be considered while designing Hydraulic Turbines.

Avoiding Cavitation

To avoid cavitation while operating Hydraulic Turbines parameters should be set such that at any point of flow static pressure may not fall below the vapor pressure of the liquid. These parameters to control cavitation are pressure head, flow rate and exit pressure of the liquid. The control parameters for cavitation free operation of hydraulic turbines can be obtained by conducting tests on model of the turbine under consideration. The parameters beyond which cavitation starts and turbine efficiency falls significantly should be avoided while operation of hydraulic turbines.

In a reaction turbine, the point of minimum pressure is usually at the outlet end of the runner blades, i.e., at the inlet to the draft tube. For the flow between such a point and the final discharge into the trail race (where the pressure is atmospheric), the Bernoulli's equation can be written, in consideration of the velocity at the discharge from draft tube to be negligibly small, as

$$\frac{p_e}{\rho g} + \frac{V_e^2}{2g} + z = \frac{p_{atm}}{\rho g} + hf \qquad (1)$$

where, *pe* and *ve* represent the static pressure and velocity of the liquid at the outlet of the runner (or at the inlet to the draft tube). The larger the value of *ve*, the smaller is the value of *pe* and the cavitation is more likely to occur. The term *hf* in Eq. (1) represents the loss of head due to friction in the draft tube and *z* is the height of the turbine runner above the tail water surface. For cavitation not to occur $pe \geq pv$, where *pv* is the vapour pressure of the liquid at the working temperature.

An important parameter in the context of cavitation is the available suction head (inclusive of both static and dynamic heads) at exit from the turbine and is usually referred to as the net positive suction head 'NPSH' which is defined as

$$NPSH = \frac{p_e}{\rho g} + \frac{V_e^2}{2g} - \frac{p_v}{\rho g} \qquad (2)$$

with the help of Eq. (1) and in consideration of negligible frictional losses in the draft tube ($hf = 0$), Eq. (2) can be written as

$$NPSH = \frac{p_{atm}}{\rho g} - \frac{p_v}{\rho g} - z \qquad (3)$$

A useful design parameter σ known as Thoma's Cavitation Parameter (after the German Engineer Dietrich Thoma, who first introduced the concept) is defined as

$$\sigma = \frac{NPSH}{H} = \frac{(p_{atm}/\rho g) - (p_v/\rho g) - z}{H} \qquad (4)$$

For a given machine, operating at its design condition, another useful parameter σ_c, known as critical cavitation parameter is define as

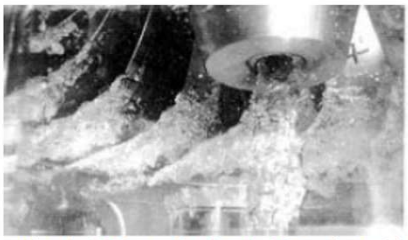
$$\sigma_c = \frac{(p_{atm}/\rho g) - (p_e/\rho g) - z}{H} \qquad (5)$$

Therefore, for cavitation not to occur $\sigma \geq \sigma_c$ (since, $pe \geq pv$)
If either z or H is increased, σ is reduced. To determine whether cavitation is likely to occur in a particular installation, the value of σ may be calculated. When the value of σ is greater than the value of σ_c for a particular design of turbine cavitation is not expected to occur.

In practice, the value of σ_c is used to determine the maximum elevation of the turbine above tail water surface for cavitation to be avoided. The parameter value of σ_c increases with an increase in the specific speed of the turbine. Hence, turbines having higher specific speed must be installed closer to the tail water level.

Note: Eq. (4) can be written as, $\sigma_c = (Ha - Hv - Hs)/H$
Hs = Turbine setting = z

Cavitation in Turbines



Traveling bubble cavitation in Francis turbine

Inlet edge cavitation in Francis turbine

Leading edge cavitation damage in Francis turbine

Critical Plant sigma values σ_c

	Francis Runners					Propeller Runners			
Specific Speed	75	150	225	300	375	375	600	750	900
σ_c	0.025	0.1	0.23	0.4	0.64	0.43	0.8	1.5	3.5

σ_c $= 0.625\ (Ns/380.78)^2$ for Francis runners

$= 0.28 + (Ns/380.78)^3 / 7.5$ for Propeller runners

$= 1.10\ (\ 0.28 + (Ns/380.78)^3 / 7.5\ )$ for Kaplan runners

Ns = Specific speed (r.p.m., kW, m)

Methods to avoid cavitation :

The following methods may be used to *avoid cavitation* :

1. Runner/turbine may be *kept under water*. But it is not advisable as the inspection and repair of the turbine is difficult. The other method to avoid cavitation zone without keeping the runner under water is *to use the runner of low specific speed.*
2. The *cavitation free runner* may be designed to fulfil the given conditions with extensive research.
3. It is possible to reduce the cavitation effect by *selecting materials which resist better the cavitation effect*. The cast steel is better than cast iron and stainless steel or alloy steel is still better than cast steel.

4. The cavitation effect can be reduced by *polishing* the surface. That is why the cast steel runners and blades are coated with stainless steel.
5. The cavitation may be avoided by selecting a runner of proper specific speed for given head.

18·16. Selection of Hydraulic Turbines

The following points should be considered while selecting right type of hydraulic turbines for hydroelectric power plant :

1. **Specific speed.** High specific speed is essential where head is low and output is large, because otherwise the rotational speed will be low which means cost of turbo-generator and power-house will be high. On the other hand, there is practically no need of chosing a high value of specific speed for high installations, because even with low specific speed high rotational speed can be attained with medium capacity plants. Refer to Table 18·2.

2. **Rotational speed.** Rotational speed depends on specific speed. Also the rotational speed of an electrical generator with which the turbine is to be directly coupled, depends on the frequency and number of pair of poles. The *value of specific speed adopted should be such that it will give the synchronous speed of the generator.*

3. **Efficiency.** The turbine selected should be such that it gives the *highest overall efficiency for various operating conditions.*

4. **Partload operation.** In general the efficiency at partloads and overloads is less than normal. For the sake of economy the turbine should always run with maximum possible efficiency to get more revenue.

When the turbine has to run at part or overload conditions *Deriaz turbine* is employed. Similarly, for low heads, Kaplan turbine will be useful for such purposes in place of propeller turbine.

5. **Cavitation.** The installation of water turbines of reaction type over the tail race is affected by *cavitation*. The critical value of cavitation factor must be obtained to see that the turbine works in *safe zone*. Such a value of cavitation factor also affects the design of turbine, especially of Kaplan, propeller and bulb types.

6. **Disposition of turbine shaft.** Experience has shown that the *vertical shaft* arrangement is better for large-sized reaction turbines, therefore, it is *almost universally adopted*. In case of *large size impulse turbines*, *horizontal shaft arrangement* is mostly employed.

7. **Head.** (*i*) *Very high heads* (*350 m and above*). For heads greater than 350 m, Pelton turbine is generally employed and there is practically no choice except in very special cases.

(*ii*) *High heads* (*150 m to 350 m*). In this range either Pelton or Francis turbine may be employed. *For higher specific speeds Francis turbine is more compact and economical than the Pelton turbine* which for the same working conditions would have to be much bigger and rather cumbersome.

(*iii*) *Medium heads* (*60 m to 150 m*). A Francis turbine is usually employed in this range. Whether a high or low specific speed unit would be used depends on the selection of the speed.

(*iv*) *Low heads* (*below 60 m*). Between 30 and 60 m heads both Francis and Kaplan turbines may be used. The latter is more expensive but yields a higher efficiency at partloads and overloads. It is therefore preferable for *variable loads. Kaplan turbine is generally employed for heads under 30 m. Propeller turbines are however, commonly used for heads up to 15 m. They are adopted only when there is practically no load variations.*

(*v*) *Very low heads.* For very low heads bulb turbines are employed these days. Although Kaplan turbines can also be used for heads from 2 m to 15 m but they are *not economical*.

Table 18·2. Criteria for Selection of Turbines

S.No.	Type of turbine	Head H(m)	Specific speed (N_s) (rpm., kW, m)	Speed ratio (K_u)	Maximum hydraulic efficiency (%)	Remarks
1.	*Pelton*: 1 jet	up to 2000	12 to 30	0·43 to 0·48	89	Employed for very high head.
	2 jets	up to 1500	17 to 50			
	4 jets	up to 500	24 to 70			
2.	*Francis*:					Full load efficiency high; partload efficiency lower than Pelton wheel.
	High-head	up to 300	80 to 150	0·6 to 0·9	93	
	Medium head	50 to 150	150 to 250			
	Low head	30 to 60	250 to 400			
3.	*Propeller and Kaplan*	4 to 60	300 to 1000	1·4 to 2	93	High part load efficiency; high discharge with low head
4.	*Bulb or tubular turbines*	3 to 10	1000 to 1200	6 to 8	91	Employed for very low head–tidal power plants.

Design procedure of the Pelton Wheel Turbine

Design outcome should be

i) Diameter of nozzle (d)
ii) Diameter of runner with buckets (D)
iii) Number of buckets(N_b)
iv) Spacing of buckets (s)
v) Width of the buckets (w)
vi) Design speed(N)
vii) Specific speed (N_s)

The following points need consideration while designing a pelton turbine.

i) Velocity of jet: the theoretical velocity of the jet is given by $V_{th} = \sqrt{2gH}$ and the actual velocity is given by $V = C_v\sqrt{2gH}$

Where H= net head on the turbine and

C_v = coefficient of velocity (the value of C_v = 0.97 to 0.99)

ii) Calculate the diameter of nozzle (d) , $Q = AV = \frac{\pi d^2}{4} \times V$

iii) Speed ratio or peripheral coefficient ($\varnothing$) : it is the ratio of the peripheral velocity of the jet bucket at their mean diameter to the theoretical velocity of the jet.

$$\varnothing = \frac{u}{\sqrt{2gH}}$$

Peripheral velocity (u) $= \frac{\pi ND}{60}$

Speed ratio has average value 0.45 to 0.47

N = rotational speed in rpm

D = diameter of the bucket or pitch diameter

iv) Jet ratio(m) : it is the ratio of the pitch circle diameter to the jet diameter

ie $m = \frac{D}{d}$

For maximum hydraulic efficiency, jet ratio lies between 11 and 15 then the diameter of the bucket can be calculated by the jet ratio

v) Number of buckets: the empirical formulae for calculating the desirable number of buckets $N_b = 0.5\frac{D}{d} + 15$

Bucket spacing $S = \frac{\pi D}{N_b}$

vi) Width of the bucket (w) it is calculated in terms of jet diameter (d)

Axial width =3d to 4d

Radial length = 2d to 3d

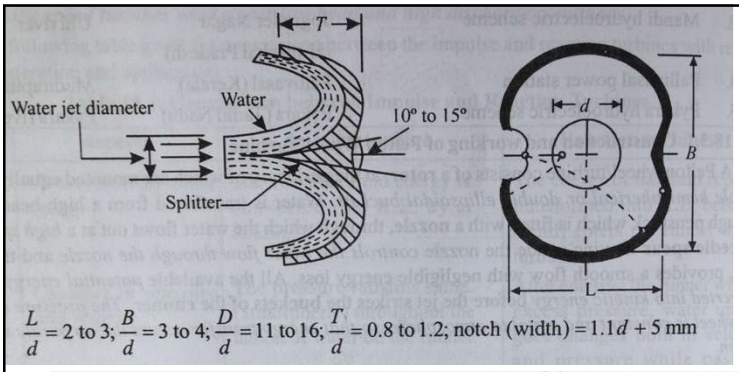
Depth = 0.8d to 1.2 d

vii) Initial rotational speed $N = \frac{60u}{\pi D}$

viii) Number of poles $= \frac{120f}{N}$, recalculate rotational speed N , using corrected number of poles

ix) Calculate power by $p = \eta\gamma QH$

x) Calculate specific speed $N_s = \frac{NP^{1/2}}{H^{5/4}}$



The angle of deflection of jet to bucket is limited to about 165º to 170º

15.6.2 Design procedure of the Francis turbine

i) calculate specific speed for the given type of the turbine

ii) calculate rotational speed of the turbine (N)

$$N_s = \frac{NP^{1/2}}{H^{5/4}}$$

iii) calculate number of poles

$$p = \frac{120f}{N}$$ where p= no of poles

iv) calculate corrected synchronous speed $N = \frac{120f}{P}$

v) calculate corrected specific speed $N_s = \frac{NP^{1/2}}{H^{5/4}}$

vi) calculate diameter of the turbine

$$D = \frac{84.6\varnothing\sqrt{H}}{N}$$

Where ∅= peripheral coefficient is given by

$\varnothing = 0.0197\ Ns^{2/3} + 0.0275$ for Francis turbine

$\varnothing = 0.0242 N_s^{2/3}$ for propeller turbine

vii) setting of turbine $Hs = H_a - H_v - \sigma H$

Numbers of turbine units

It is most effective to have a minimum number of units at a given installation. Increase in turbine size increases the efficiency of the plant. However, multiple units may be necessary to make the most efficient use of water where flow variation is great. Factors as space limitation by geological characteristics of existing structure may dictate large or small units. The difficulty in transportation or large runners sometimes makes it necessary to limit their size. Isolation system may require more number of units. The percentage of load covered by the plant is another important factor if the contribution of the plant to system is more; the number of units should be more. For the power plant it is better to make the units are identical also somehow dictates the number of plants. If the power generation from the plant is fluctuating, the number of units will be increased and vice versa.

3.2. Work done and Efficiency of a Pelton Wheel

Fig. 2.5 shows the velocity triangles.

Let,

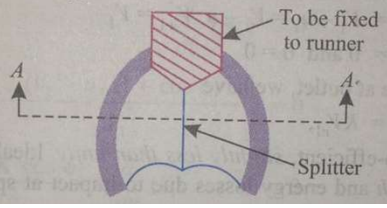
N = Speed of wheel in r.p.m.,

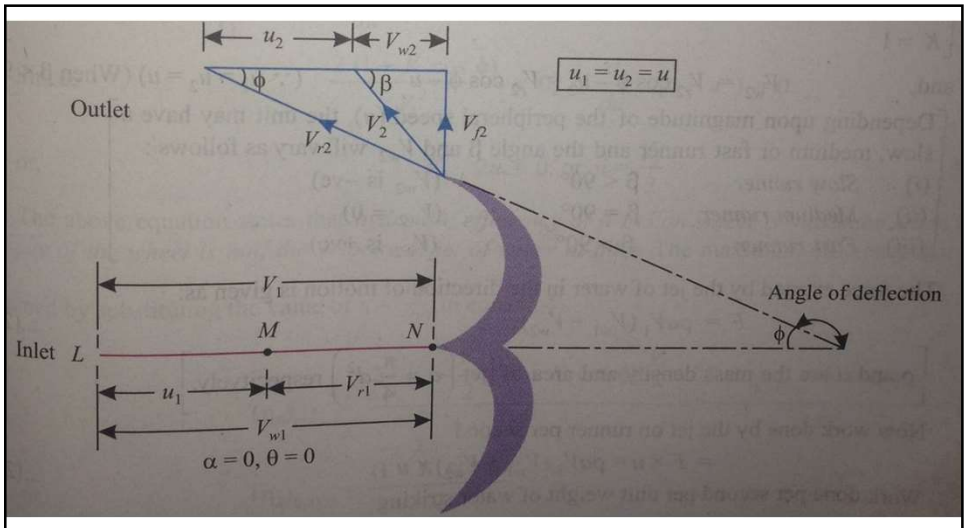
D = Diameter of the wheel,

d = Diameter of the jet,

u = Peripheral (or circumferential) velocity of runner. It will be same at inlet and outlet of the runners at the mean pitch. (*i.e.* $u = u_1 = u_2$)

$$= \frac{\pi DN}{60},$$





V_1 = Absolute velocity of water at inlet,

V_{r1} = Jet velocity relative to vane/bucket at inlet,

α = Angle between the direction of the jet and direction of motion of the vane/bucket (also called *guide angle*),

θ = Angle made by the relative velocity (V_{r1}) with the direction of motion at inlet (also called *vane angle at inlet*),

V_{w1} and V_{f1} = The components of the velocity of the jet V_1, in direction of motion and perpendicular to the direction of motion of the vane respectively;

V_{w1} is also known as *velocity of whirl* at inlet,

V_{f1} is also known as *velocity of flow* at inlet,

V_2 = Velocity of jet, leaving the vane or velocity of jet at outlet of the vane,

V_{r2} = Relative velocity of the jet with respect to the vane at outlet,

ϕ = Angle made by the relative velocity V_{r2} with the direction of motion of the vane at outlet and also called *vane angle at outlet*,

β = Angle made by the velocity V_2 with the direction of motion of the vane at outlet, and

V_{w2} and V_{f2} = Components of the velocity V_2, in the direction of motion of vane and perpendicular to the direction of motion of vane at outlet;

V_{w2} is also called the *velocity of whirl at outlet*, and

V_{f2} is also called the *velocity of flow at outlet*.

Inlet. The velocity triangle at *inlet* will be a *straight line* where

$$V_{r1} = V_1 - u_1 = V_1 - u, \; V_{w1} = V_1 \qquad (\because u_1 = u_2 = u)$$

$$\alpha = 0 \text{ and } \theta = 0$$

Outlet : From velocity triangle at outlet, we have

$$V_{r2} = KV_{r1},$$

[where, K = blade friction co-efficient, *slightly less than unity*. Idealy when bucket surfaces are *perfectly smooth* and energy losses due to impact at splitter are *neglected*, $K = 1$]

and, $$V_{w2} = V_{r2}\cos\phi - u_2 = V_{r2}\cos\phi - u \qquad (\because u_1 = u_2 = u)$$

The force exerted by the jet of water in the direction of motion is given as:

$$F = \rho a V_1 (V_{w1} + V_{w2}) \quad ...(2.1)$$

[ρ and a are the mass density and area of jet $\left(a = \frac{\pi}{4} d^2\right)$ respecitvely.]

Now work done by the jet on runner per second

$$= F \times u = \rho a V_1 (V_{w1} + V_{w2}) \times u \quad ...(2.2)$$

Work done per second per unit weight of water striking

$$= \frac{\rho a V_1 (V_{w1} + V_{w2}) \times u}{\text{Weight of water striking}} = \frac{\rho a V_1 (V_{w1} + V_{w2}) \times u}{\rho a V_1 \times g}$$

$$= \frac{1}{g} (V_{w1} + V_{w2}) u \quad ...[2.2\ (a)]$$

The energy supplied to the jet at inlet is in the form of K.E. and is equal to $\frac{1}{2} m V_1^2$.

$\therefore$ Kinetic energy (K.E.) of jet per second $= \frac{1}{2} (\rho a V_1) \times V_1^2$

$\therefore$ Hydraulic efficiency, $\eta_h = \frac{\text{Work done per second}}{\text{K.E. of jet per second}} = \frac{\rho a V_1 (V_{w1} + V_{w2}) \times u}{\frac{1}{2} (\rho a V_1) \times V_1^2}$

or, $$\eta_h = \frac{2 (V_{w1} + V_{w2}) \times u}{V_1^2} \quad ...(2.3)$$

From inlet and outlet velocity triangles, we have:

$$V_{w1} = V_1,\ V_{r1} = V_1 - u_1 = V_1 - u$$

$$V_{w2} = V_{r2} \cos \phi - u_2 = V_{r2} \cos \phi - u = K V_{r1} \cos \phi - u = K (V_1 - u) \cos \phi - u$$

Substituting the values of V_{w1} and V_{w2} in eqn (2.3), we have:

$$\eta_h = \frac{2[V_1 + K (V - u) \cos \phi - u] u}{V_1^2} = \frac{2[(V_1 - u)(1 + K \cos \phi)] u}{V_1^2} \quad ...(2.4)$$

The hydraulic efficiency will be *maximum* for given value of V_1 when,

$$\frac{d}{du} (\eta_h) = 0$$

i.e., $$\frac{d}{du} \left[\frac{2 (V_1 - u)(1 + K \cos \phi) u}{V_1^2}\right] = 0$$

or, $$\frac{2 (1 + K \cos \phi)}{V_1^2} \times \frac{d}{du} (V_1 u - u^2) = 0$$

Since, $$\frac{2 (1 + K \cos \phi)}{V_1^2} \neq 0,\ \therefore \frac{d}{du} (V_1 u - u^2) = 0$$

or, $$V_1 - 2u = 0,\ \text{or},\ u = \frac{V_1}{2} \quad ...(2.5)$$

The above equation states that *hydraulic efficiency of a Pelton wheel is maximum when the velocity of the wheel is half the velocity of jet of water at inlet.* The maximum efficiency can be

obtained by substituting the value of $u = \frac{V_1}{2}$ in eqn. (2.4).

$$(\eta_h)_{max} = \frac{2\left(V_1 - \frac{V_1}{2}\right)(1 + K\cos\phi)\frac{V_1}{2}}{V_1^2} = \frac{2 \times \frac{V_1}{2}(1 + K\cos\phi) \times \frac{V_1}{2}}{V_1^2}$$

or,
$$(\eta_h)_{max} = \frac{(1 + K\cos\phi)}{2} \quad ...(2.6)$$

If friction factor, $K = 1$ (*i.e.*, assuming *no friction*), we have

$$(\eta_h)_{max} = \frac{1 + \cos\phi}{2} \quad ...[2.6(a)]$$

Example 2.1. *A Pelton wheel is receiving water from a penstock with a gross head of 510 m. One-third of gross head is lost in friction in the penstock. The rate of flow through the nozzle fitted at the end of the penstock is 2·2 m^3/s. The angle of deflection of the jet is 165°. Determine :*

(i) The power given by water to the runner, and

(ii) Hydraulic efficiency of the Pelton wheel.

Take C_v (co-efficient of velocity) = 1·0 and speed ratio = 0·45.

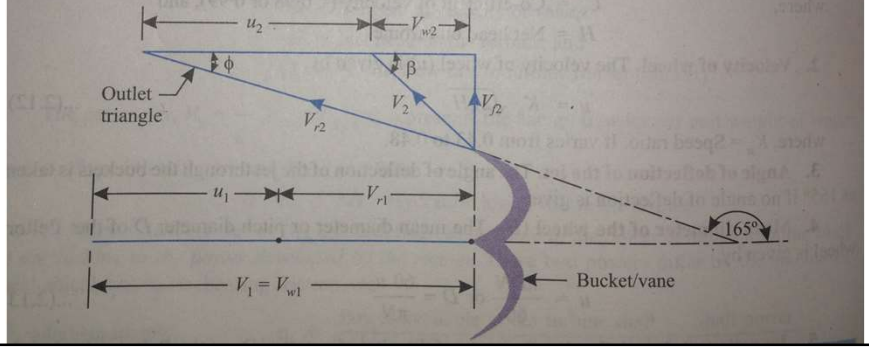
Solution. Gross head, $H_g = 510$ m

Head lost in friction, $h_f = \frac{H_g}{3} = \frac{510}{3} = 170$ m

$\therefore$ Net head, $H = H_g - h_f = 510 - 170 = 340$ m

Discharge, $Q = 2{\cdot}2$ m^3/s

Angle of deflection = 165°



∴ Angle, $\phi = 180° - 165° = 15°$

Co-efficient of velocity, $C_v = 1.0$

Speed ratio, $K_u = 0.45$

(i) The power given by water to the runner :

Velocity of jet, $V_1 = C_v\sqrt{2gH} = 1.0\sqrt{2 \times 9.81 \times 340} = 81.67$ m/s

Velocity of wheel, $u = K_u\sqrt{2gH} = 0.45\sqrt{2 \times 9.81 \times 340} = 36.75$ m/s

Refer to fig. 2.7. $V_{r1} = V_1 - u_1 = V_1 - u = 81.67 - 36.75 = 44.92$ m/s $(\because u_1 = u_2 = u)$

Also, $V_{w1} = V_1 = 81.67$ m/s

From outlet velocity triangle, we have:

$V_{r2} = V_{r1} = 44.92$ m/s

Also, $V_{r2}\cos\phi = u_2 + V_{w2} = u + V_{w2}$

or, $V_{w2} = V_{r2}\cos\phi - u = 44.92 \cos 15° - 36.75 = 6.64$ m/s

Work done by the jet on the runner per second ...[Eqn (2.2)]

$= \rho Q (V_{w1} + V_{w2}) \times u$

$= 1000 \times 2.2 (81.67 + 6.64) \times 36.75 = 7139863$ Nm/s

∴ Power given by water to the runner = 7139863 J/s

or, $W \simeq$ **7139.8 kW (Ans.)**

(ii) Hydraulic efficiency of the Pelton wheel, η_h :

$$\eta_h = \frac{2 (V_{w1} + V_{w2}) \times u}{V_1^2} \quad \text{...[Eqn (2.3)]}$$

$$= \frac{2 (81.67 + 6.64) \times 36.75}{(81.67)^2} = 0.973 \quad \text{or} \quad \textbf{97.3 \% (Ans.)}$$

Alternatively :

$$\eta_h = \frac{(V_{w1} + V_{w2})\, u}{gH} \quad \text{...[Eqn (2.7)]}$$

$$= \frac{(18.67 + 6.64) \times 36.75}{9.81 \times 340} = 0.973 \quad \text{or} \quad \textbf{97.3 \% (Ans.)}$$

Example 2.2. *A Pelton wheel having a mean bucket diameter of 1·2 m is running at 1000 r.p.m. The net head on the Pelton wheel is 840 m. If the side clearance angle is 15° and discharge through the nozzle is 0·12 m^3/s, determine :*

(i) Power available at the nozzle, and

(ii) Hydraulic efficiency of the turbine.

Solution. Mean diameter of Pelton wheel, $D = 1.2$ m

Speed of wheel, $N = 1000$ r.p.m.

$$\therefore \quad \text{Tangential velocity of the wheel, } u = \frac{\pi DN}{60} = \frac{\pi \times 1.2 \times 1000}{60} = 62.83 \text{ m/s}$$

Net head on the turbine, $H = 840$ m

Side clearance angle, $\theta = 15°$

Discharge $Q = 0.12$ m^3/s

(i) Power available at the nozzle :

Velocity of jet at inlet, $V_1 = C_v\sqrt{2gH}$

$= 1.0\sqrt{2 \times 9.81 \times 840} = 128.38$ m/s

(Assume $C_v = 1.0$ if not given)

$\therefore$ Power available at nozzle

$= wQH = 9810 \times 0.12 \times 840 = 988848$ Nm/s or J/s or W $\simeq$ **988.85 kW (Ans.)**

(ii) Hydraulic efficiency, η_h :

$$\eta_h = \frac{2(V_1 - u)(1 + K\cos\phi)u}{V_1^2} \quad \text{...[Eqn. (2.4)]}$$

$$= \frac{2(128.38 - 62.83)(1 + \cos 15°) \times 62.83}{(128.38)^2} = \frac{131.1(1 + 0.966) \times 62.83}{(128.38)^2}$$

(Assume $K = 1$)

$= 0.982$ or **98.2 % (Ans.)**

Example 2.3. *A Pelton wheel is to be designed for the following specifications :*

Power (brake or shaft) ... *9560 kW*
Head ... *350 metres*
Speed ... *750 r.p.m.*
Overall efficiency ... *85%*
Jet diameter ... *not to exceed 1/6 th of the wheel diameter*

Determine the following :

(i) The wheel diameter, *(ii) Diameter of the jet, and*
(iii) The number of jets required.

Take $C_v = 0{\cdot}985$, Speed ratio = 0.45.

Solution. Shaft or brake power = 9560 kW
Head, H = 350 m
Speed, N = 750 r.p.m.
Overall efficiency, η_0 = 85%
Ratio of jet diameter to wheel, $\frac{d}{D} = \frac{1}{6}$
Co-efficient of velocity, C_v = 0.985
Speed ratio, K_u = 0.45

(i) The wheel diameter, D :

Velocity of jet, $V_1 = C_v\sqrt{2gH} = 0.985\sqrt{2 \times 9.81 \times 350} = 81.62$ m/s

The velocity of wheel, $u = u_1 = u_2$

$= K_u \times \sqrt{2gH} = 0.45\sqrt{2 \times 9.81 \times 350} = 37.3$ m/s

But, $u = \frac{\pi DN}{60}$

$\therefore \quad 37.3 = \frac{\pi D \times 750}{60}$, or, $D = \frac{37.3 \times 60}{\pi \times 750} = \mathbf{0.95\ m}$ **(Ans.)**

(ii) Diameter of the jet, d :

$$\frac{d}{D} = \frac{1}{6}$$

$$\therefore \quad d = \frac{D}{6} = \frac{0.95}{6} = \mathbf{0.158\ m}\ \textbf{(Ans.)}$$

(iii) The number of jets required :

Discharge of one jet, q = Area of jet × velocity of jet

$$= \frac{\pi}{4} d^2 \times V_1 = \frac{\pi}{4} \times 0.158^2 \times 81.62 = 1.6\ \text{m}^3/\text{s}$$

Now, overall efficiency, $\eta_0 = \dfrac{\text{Shaft power}}{\text{Water power}} = \dfrac{9560}{wQH}$

or, $$0.85 = \frac{9560}{9.81 \times Q \times 350} \qquad (\because w = 9.81\ \text{kN/m}^3)$$

$$\therefore \quad \text{Total discharge, } Q = \frac{9560}{0.85 \times 9.81 \times 350} = 3.27\ \text{m}^3/\text{s}$$

$$\therefore \quad \text{Number of jets} = \frac{\text{Total discharge}}{\text{Discharge of one jet}} = \frac{Q}{q} = \frac{3.27}{1.6} = \mathbf{2\ jets}\ \textbf{(Ans.)}$$

4.1.1. Work done and efficiency of Francis turbine

Net head at the turbine runner : In the Fig. 2.12,

H_g = Gross head = Difference of water levels between head race and tail race;

h_f = Loss of head in the penstock;

H = Net head = $(H_g - h_f)$. The net head is also called *available* or *working* or *operation head*.

Also, $$H = \begin{bmatrix}\text{Total energy available at exit}\\ \text{from the penstock}\end{bmatrix} - \begin{bmatrix}\text{total energy available at exit}\\ \text{from the draft tube}\end{bmatrix}$$

$$= \left(\frac{p}{w} + \frac{V^2}{2g} + z\right)_{\text{penstock}} - \left(\frac{p}{w} + \frac{V^2}{2g} + z\right)_{\text{draft tube}}$$

If the draft tube exit is at tail race level, and the datum is also taken at that level, then,

$$H = \left(\frac{p}{w} + \frac{V^2}{2g} + z\right)_{\text{penstock}} - \frac{V_d^2}{2g}$$

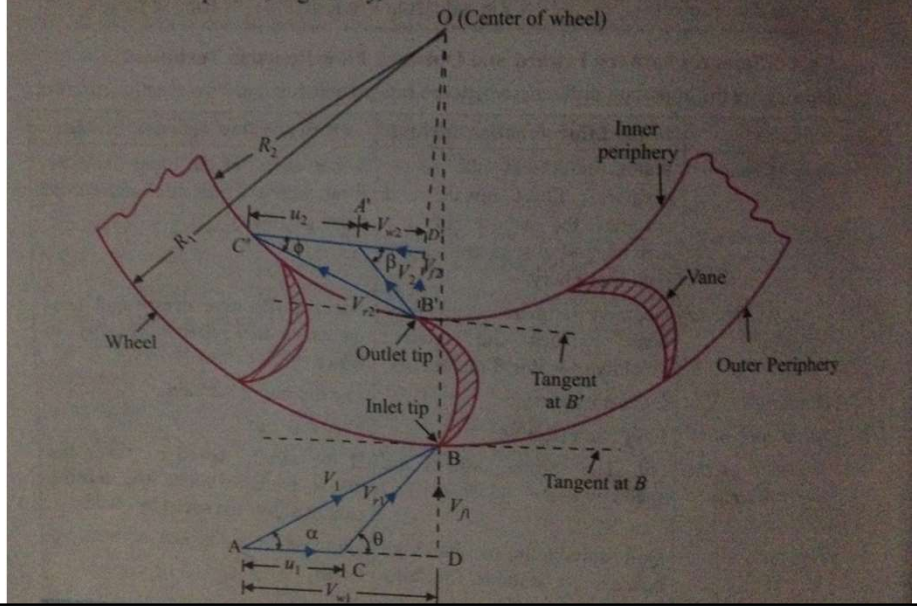
(where, V_d = velocity at the exit of the draft tube)

Neglecting the velocity at the draft tube exit (V_d), we have:

$$H = \left(\frac{p}{w} + \frac{V^2}{2g} + z\right) \qquad \ldots(2.16)$$

Work done by the runner :

Fig. 2.13 shows the runner and the velocity diagrams (inlet and outlet) for an inward flow reaction turbine. The general expression for the work done with usual notations according to the Euler momentum equation, is given by,



$$\text{Work done} = \rho Q\,(V_{w1}\,u_1 - V_{w2}\,u_2)$$

$$= \frac{wQ}{g}\,(V_{w1}\,u_1 - V_{w2}\,u_2) \qquad \ldots(2.17)$$

where, Q = Discharge through the runner, m^3/s.

The maximum output under given conditions is obtained when $V_{w2} = 0$.

Thus, the maximum work done is given by,

$$\text{Work done} = \frac{wQ}{g}\,(V_{w1}\,u_1) \qquad \ldots(2.18)$$

This *discharge in this case is radial*. For radial discharge, the *absolute velocity at exit is radial.*

Hydraulic efficiency, η_h :

If H is the net head, then input to the turbine = wQH.

$$\eta_h = \frac{\text{Power developed by the runner}}{\text{Power supplied to the turbine (water power)}} = \frac{\frac{wQ}{g}\,(V_{w1}\,u_1)}{wQH}$$

or, $$\eta_h = \frac{V_{w1}\,u_1}{gH} \qquad \ldots(2.19)$$

However, if the velocity of whirl at the exit is *not* zero, then

$$\eta_h = \frac{V_{w1}\, u_1 - V_{w2}\, u_2}{gH} \qquad ...[2.19(a)]$$

The hydraulic efficiency of the Francis turbine varies from 85 to 90 percent.

Mechanical efficiency, η_m :

The mechanical efficiency is given by:

$$\eta_m = \frac{\text{Shaft power (P)}}{\text{Power developed by the runner}} \qquad ...(2.20)$$

Overall efficiency, η_0 :

The overall efficiency is given as:

$$\eta_0 = \frac{\text{Shaft water}}{\text{Water power}} = \frac{P}{wQH} \qquad ...(2.21)$$

and,

$$\eta_0 = \eta_h \times \eta_m \qquad [2.21(a)]$$

The overall efficiency varies from 80 to 90 percent.

4.1.2. Working proportions of a Francis turbine

The following working proportions pertain to a Francis turbine :

1. **Ratio of width to diameter $\left(\frac{B}{D}\right)$:**

The ratio of width (B_1) to the diameter of the wheel (D_1) at inlet is represented by n. Thus,

$$n = \frac{B_1}{D_1}$$

The value of n varies from 0.10 to 0.45.

2. **Flow ratio (K_f) :**

Flow ratio is the *ratio of the velocity of flow at inlet to the theoretical jet velocity*. Thus,

$$\text{Flow ratio, } K_f = \frac{V_{f1}}{\sqrt{2gH}} \qquad ...(2.23)$$

The value of K_f varies from 0.15 to 0.30.

3. **Speed ratio (K_u)**

Speed ratio is the ratio of the peripheral speed at inlet to the theoretical jet velocity. Thus,

$$\text{Speed ratio, } K_u = \frac{u}{\sqrt{2gH}}$$

The value of K_u ranges from 0.6 to 0.9.

2.4.1.3. Design of Francis turbine runner

The runner of a Francis turbine is required to be designed to develop a known power P, when running at a known speed N r.p.m. under a known head H. The design of the runner involves the determination of its *size* and the *vane angles*.

The design of a Francis turbine runner is carried out as follows :

1. Assume suitable values of η_0, η_h, n, K_f and K_r.
2. Determine the required discharged Q from the relation:

$$P = \eta_0 \times wQH \qquad ...(2.24)$$

3. Obtain the velocity of flow from the discharge and flow area.

Let B_1, D_1 and t_1 respectively be the width, diameter and thickness of runner vane at inlet (Fig. 2.14).

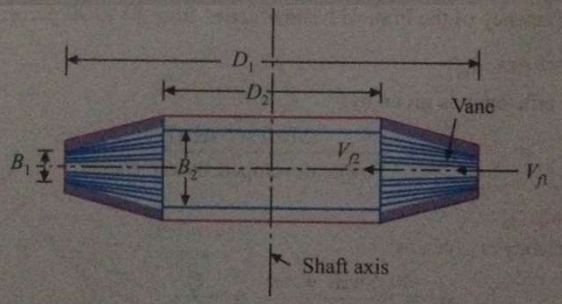


Fig. 2.14. Entry of flow to runner vane.

Then, total area at the outer periphery (*i.e.*, at the runner inlet),

$$A = (\pi D_1 * K_{t1})\, B_1 = K_{t1} \pi D_1 B_1 \qquad ...(2.25)$$

where K_{t1} is known as *vane thickness factor/co-efficient*; its value is *always less than unity* (usually of the order of 0.95 or so).

Discharge = Area of flow × velocity of flow

i.e.,
$$Q = K_{t1} \pi D_1 B_1 \times V_{f1} \qquad ...(2.26)$$

$$\therefore \quad \text{Flow velocity, } V_{f1} = \frac{Q}{K_{t1} \pi D_1 B_1} = \frac{Q}{K_t \pi n D_1^2} \quad (\because B_1 = nD_1) \qquad ...[2.26(a)]$$

Also,
$$V_{f1} = K_f \sqrt{2gH}, \quad \therefore \quad K_f \sqrt{2gH} = \frac{Q}{K_t \pi n D_1^2}$$

or,
$$D_1 = \left[\frac{Q}{(K_f \sqrt{2gH})\, K_t \pi n} \right]^{1/2} \qquad ...(2.27)$$

Then, width $B_1 = nD_1$ [Eqn. (2.22)]

4. Find the rim velocity (tangential velocity) u_1 from the relation:

$$u_1 = \frac{\pi D_1 N}{60}$$

5. Find the velocity of whirl at inlet (V_{w1}) from Eq [2.19)

$$\eta_h = \frac{V_{w1}\, u_1}{gH}, \quad \text{or,} \quad V_{w1} = \frac{\eta_h gH}{u_1} \qquad ...(2.28)$$

6. Obtain the guide vane angle (α) and the runner vane angle (θ) from the following relations obtained from inlet velocity triangle (Fig. 2.13):

$$\tan \alpha = \frac{V_{f1}}{V_{w1}} \qquad ...(2.29)$$

and,
$$\tan \theta = \frac{V_{f1}}{V_{w1} - u_1} \qquad ...(2.30)$$

7. Assume runner diameter D_2 at the outlet to be approximately one-half the diameter at inlet.

Thus, $D_2 = \frac{D_1}{2}$ and $u_2 = \frac{u_1}{2}$.

8. The velocity of flow at the exit (V_{f2}) is obtained as follows :

$$Q = K_{t1}\pi D_1 B_1 V_{f1} = K_{t2}\pi D_2 B_2 V_{f2} \qquad ...Continuity\ equation$$

Thus, $$\frac{V_{f1}}{V_{f2}} = \frac{K_{t2}\pi D_2 B_2}{K_{t1}\pi D_1 B_1}$$

Usually, it is presumed that $V_{f1} = V_{f2}$ and $K_{t1} = k_{t2}$, that gives $B_2 = 2B_1$

9. Find the runner vane angle at exit (ϕ) from the velocity outlet triangle, assuming the discharge at the runner exit to be radial ($\beta = 90°$). Thus,

$$\tan \theta = \frac{V_{f2}}{u_2} \qquad ...(2.31)$$

10. The number of vanes varies from 16 to 24. In order to *avoid periodic impulse, the number of vanes should be either one more or one less than the number of guide vanes.*

Example 2.16. *An inward flow reaction turbine has external and internal diameters as 1·08 m and 0.54 m. The turbine is running at 200 r.p.m. The width of the turbine at inlet is 240 mm and velocity of flow through the runner is constant and is equal to 2.16 m/s. The guide blades make an angle of 10° to the tangent of the wheel and discharge at the outlet of the turbine is radial. Draw the inlet and outlet velocity triangles and determine :*

- **(i)** *The absolute velocity of water at inlet of the runner,*
- **(ii)** *The velocity of whirl at inlet,*
- **(iii)** *The relative velocity at inlet,*
- **(iv)** *The runner blade angles,*
- **(v)** *Width of runner at outlet,*
- **(vi)** *Weight of water flowing through the runner per second,*
- **(vii)** *Head at inlet of the turbine,*
- **(viii)** *Power developed, and*
- **(ix)** *Hydraulic efficiency of the turbine.*

Solution.

External diameter, $D_1 = 1.08$ m
Internal diameter, $D_2 = 0.54$ m
Speed, $N = 200$ r.p.m.
Width at inlet, $B_1 = 240$ mm $= 0.24$ m
Velocity of flow, $V_{f1} = V_{f2} = 2.16$ m/s
Guide blade angle, $\alpha = 10°$

Discharge at outlet .... radial

$\therefore \quad \beta = 90°$ and $V_{w2} = 0$

Tangential velocity of wheel at inlet, $u_1 = \dfrac{\pi D_1 N}{60} = \dfrac{\pi \times 1.08 \times 200}{60} = 11.31$ m/s

Tangential velocity of wheel at outlet, $u_2 = \dfrac{\pi D_2 N}{60} = \dfrac{\pi \times 0.54 \times 200}{60} = 5.65$ m/s

(*i*) Absolute velocity of water at inlet of the runner, V_1 :

From inlet velocity triangle, we have:

$$V_1 \sin \alpha = V_{f1}$$

$$\therefore \quad V_1 = \frac{V_{f1}}{\sin \alpha} = \frac{2.16}{\sin 10°} = \mathbf{12.44\ m/s\ (Ans.)}$$

(*ii*) Velocity of whirl at inlet, V_{w1} :

$$V_{w1} = V_1 \cos \alpha = 12.44 \times \cos 10° = \mathbf{12.25\ m/s\ (Ans.)}$$

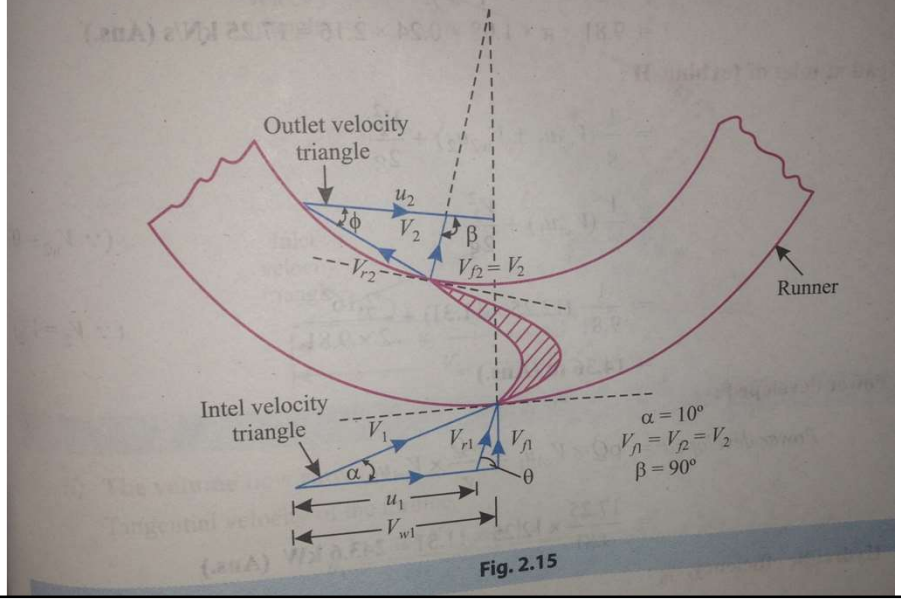


Fig. 2.15

(iii) Relative velocity at inlet, V_{r1} :

$$V_{r1} = \sqrt{(V_{w1} - u_1)^2 + V_{f1}^2}$$

$$= \sqrt{(12{\cdot}25 - 11.31)^2 + (2.16)^2} = \mathbf{2.35\ m/s\ (Ans.)}$$

(iv) Runner blade angles, θ, ϕ :

Again, from inlet velocity triangle, we have:

$$\tan\theta = \frac{V_{f1}}{V_{w1} - u_1} = \frac{2.16}{12.25 - 11.31} = 2.298$$

$$\therefore \quad \theta = \tan^{-1} 2.298 = \mathbf{66.48°\ (Ans.)}$$

From the outlet velocity triangle, we have:

$$\tan\theta = \frac{V_{f2}}{u_2} = \frac{2.16}{5.65} = 0.382$$

$$\therefore \quad \phi = \tan^{-1} 0.382 = \mathbf{20.9°\ (Ans.)}$$

(v) Width of runner at outlet, B_2 :

$$\pi D_1 B_1 V_{f1} = \pi D_2 B_2 V_{f2} (= Q) \qquad \textit{...Continuity equation}$$

or,
$$D_1 B_1 = D_2 B_2 \qquad (\because V_{f1} = V_{f2})$$

$$\therefore \quad B_2 = \frac{D_1 B_1}{D_2} = \frac{1.08 \times 0.24}{0.54} = 0.48 \text{ m or } \mathbf{480\ mm\ (Ans.)}$$

(vi) Weight of water flowing through the runner per second :

Weight of water flowing per second

$$= wQ = w \times \pi D_1 B_1 V_{f1}$$

$$= 9.81 \times \pi \times 1.08 \times 0.24 \times 2.16 = \mathbf{17.25\ kN/s\ (Ans.)}$$

(vii) Head at inlet of turbine, H :

$$= \frac{1}{g}(V_{w1}u_1 \pm V_{w2}u_2) + \frac{V_2^2}{2g}$$

$$= \frac{1}{g}(V_{w1}u_1) + \frac{V_2^2}{2g} \qquad (\because V_{w2} = 0)$$

$$= \frac{1}{9.81}(12.25 \times 11.31) + \frac{2.16^2}{2 \times 9.81} \qquad (\because V_2 = V_{f2})$$

$$= \mathbf{14.36\ m\ (Ans.)}$$

(viii) Power developed :

$$\textit{Power developed} = \rho Q \times V_{w1}u_1 = \frac{wQ}{g} \times V_{w1}u_1$$

$$= \frac{17.25}{9.81} \times 12.25 \times 11.31 = \mathbf{243.6\ kW\ (Ans.)}$$

(ix) Hydraulic efficiency, η_h :

$$\eta_h = \frac{V_{w1} u_1}{gH} \quad \text{...[Eqn (2.19)]}$$

$$= \frac{12.25 \times 11.31}{9.81 \times 14.36} = 0.9835 \quad \text{or} \quad \mathbf{98.35\%\ (Ans.)}$$

Example 2.17. *A reaction turbine works at 450 r.p.m. under a head of 120 m. Its diameter at inlet is 1.2 m and the flow area is 0.4 m^2 . The angles made by absolute and relative velocities at inlet are 20° and 60° respectively with the tangential velocity. Determine :*

(i) The volume flow rate,

(ii) The power developed, and

(iii) The hydraulic efficiency. **[PTU]**

Solution.

Speed of turbine, N = 450 r.p.m

Head, H = 120 m

Diameter at inlet, D_1 = 1.2 m

Flow area, $\pi D_1 B_1$ = 0.4 m^2

Angle made by absolute velocity, α = 20°

Angle made by the relative velocity at inlet, θ = 60°

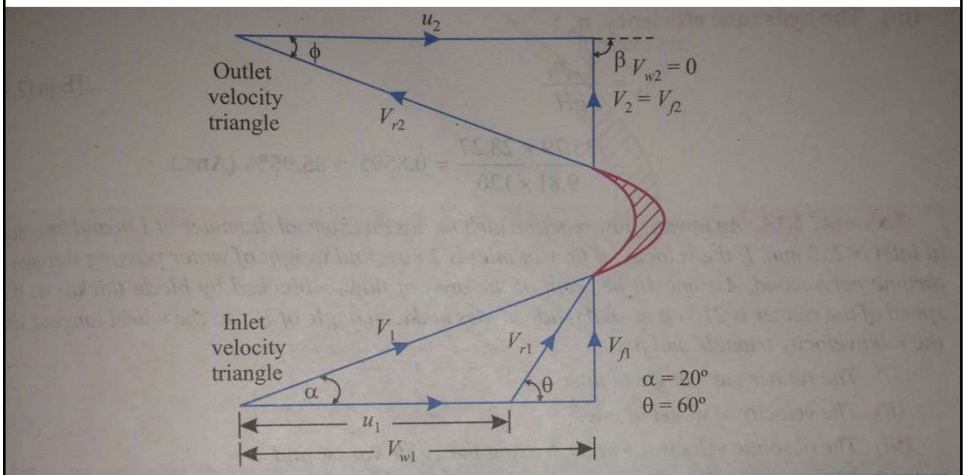


Fig. 2.16

(i) The volume flow rate, Q :

Tangential velocity of the turbine,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 1.2 \times 450}{60} = 28.27 \text{ m/s}$$

From inlet velocity triangle, we have:

$$\tan \alpha = \frac{V_{f1}}{V_{w1}}, \quad \text{or,} \quad \tan 20° = \frac{V_{f1}}{V_{w1}}$$

$$\therefore \quad V_{f1} = V_{w1} \tan 20° = 0.364\, V_{w1} \qquad ...(i)$$

Also,
$$\tan \theta = \frac{V_{f1}}{V_{w1} - u_1} = \frac{0.364\, V_{w1}}{V_{w1} - 28.27} \qquad (\because V_{f1} = 0.364\, V_{w1})$$

or,
$$\tan 60° = \frac{0.364\, V_{w1}}{V_{w1} - 28.27}, \text{ or, } 1.732 = \frac{0.364\, V_{w1}}{V_{w1} - 28.27}$$

or, $1.732\,(V_{w1} - 28.27) = 0.364\, V_{w1}$, or, $1.732\, V_{w1} - 48.96 = 0.364\, V_{w1}$

$$\therefore \quad V_{w1} = \frac{48.96}{(1.732 - 0.364)} = 35.79 \text{ m/s}$$

From eqn. (*i*), we have:

$$V_{f1} = 0.364 \times 35.79 = 13.027 \text{ m/s}$$

$\therefore$ Volume flow rate, $Q = \pi D_1 B_1 \times V_{f1}$

But, $\pi D_1 B_1 = 0.4 \text{ m}^2$ ...(*Given*)

$$\therefore \quad Q = 0.4 \times 13.027 = \mathbf{5.211\ m^3/s\ (Ans.)}$$

(*ii*) Power developed :

Work done per second $= \rho Q\,(V_{w1} u_1)$ $[\because V_{w2} = 0$ ...Given$]$

$= 1000 \times 5.211 \times 35.79 \times 28.27 = 5272402$ Nm/s or J/s

$\therefore$ Power developed $= 5272402$ J/s, or, $W = \mathbf{5272.4\ kW\ (Ans.)}$

(*iii*) The hydraulic efficiency, η_h :

$$\eta_h = \frac{V_{w1} u_1}{gH} \qquad ...[\text{Eqn (2.19)}]$$

$$= \frac{35.79 \times 28.27}{9.81 \times 120} = 0.8595 = \mathbf{85.95\%\ (Ans.)}$$

Example 2.18. *An inward flow reaction turbine has an external diameter of 1 m and its breadth at inlet is 250 mm. If the velocity of flow at inlet is 2 m/s, find weight of water passing through the turbine per second. Assume 10 per cent of the area of flow is blocked by blade thickness. If the speed of the runner is 210 r.p.m. and guide blades make an angle of 10° to the wheel tangent, draw the inlet velocity triangle and find :*

(*i*) *The runner vane angle at inlet,*

(*ii*) *The velocity of wheel at inlet,*

(*iii*) *The absolute velocity of water leaving the guide vanes, and*

(*iv*) *The relative velocity of water entering the runner blade.* **[Rookee University]**

Solution.

External diameter, $D_1 = 1$ m

Breadth of inlet, $B_1 = 250$ m $= 0{\cdot}25$ m

Velocity of flow at inlet, $V_{f1} = 2.0$ m/s

Area of flow blocked by blade thickness = 10 %

Speed of the runner, $N = 210$ r.p.m.

Guide blade angle, $\alpha = 10°$

Weight of water passing through the turbine :

$$\text{Area blocked by vane thickness} = \frac{10}{100} \times \pi D_1 B_1 = 0.1\,\pi D_1 B_1$$

$\therefore$ Actual area through which flow takes place,

$$A = \pi D_1 B_1 - 0.1\,\pi D_1 B_1 = 0.9\,\pi D_1 B_1$$
$$= 0.9\,\pi \times 1 \times 0.25 = 0.7068 \text{ m}^2$$

$\therefore$ Weight of water passing per second through the turbine

$$= w \times A \times V_{f1} = 9.81 \times 0.7068 \times 2$$
$$= \mathbf{13.68 \text{ kN/s (Ans.)}}$$

Inlet velocity triangle is shown in Fig 2.17.

(*i*) **The runner vane angle at inlet, θ :**

Tangential velocity of wheel at inlet,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 1 \times 10}{60} = 10.99 \text{ m/s}$$

From inlet velocity triangle, we have:

$$\tan \alpha = \frac{V_{f1}}{V_{w1}} = \frac{2.0}{V_{w1}}, \text{ or, } V_{w1} = \frac{2.0}{\tan \alpha} = \frac{2.0}{\tan 10°} = 11.34 \text{ m/s}$$

$$\tan \theta = \frac{V_{f1}}{V_{f1} - u_1} = \frac{2.0}{11.34 - 10.99} = 5.714$$

$$\theta = \tan^{-1} 5.714$$
$$= \mathbf{80.07° \text{ (Ans.)}}$$

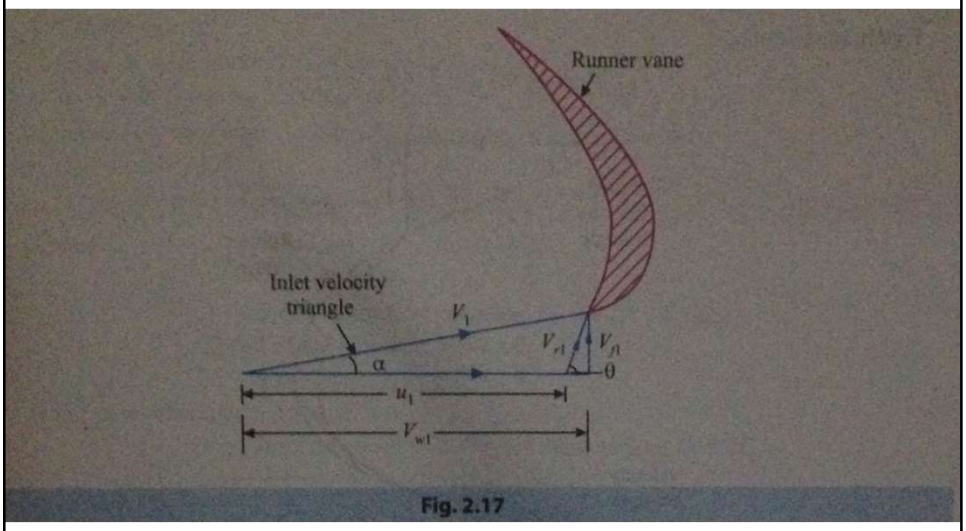


Fig. 2.17

(ii) The velocity of wheel at inlet u_1 :

The velocity of wheel at inlet = u_1 = **10.99 m/s (Ans.)**

(iii) The absolute velocity of water leaving the guide vanes, V_1 :

Again, from inlet velocity triangle , we have:

$$\sin\alpha = \frac{V_{f1}}{V_1}, \quad \text{or,} \quad \sin 10° = \frac{2.0}{V_1}$$

$$\therefore \quad V_1 = \frac{2.0}{\sin 10°} = \textbf{11.52 m/s (Ans.)}$$

(iv) The relative velocity of water entering the runner blade, V_{r1} :

$$\sin\theta = \frac{V_{f1}}{V_{r1}}, \quad \text{or,} \quad \sin 80.07° = \frac{2.0}{V_{r1}}$$

$$\therefore \quad V_{r1} = \frac{2.0}{\sin 80.07°} = \textbf{2.03 m/s (Ans.)}$$

Governing of Hydraulic Turbines

- N = 120f/p implies speed of the generator can be maintained at a constant level only when the speed of the turbine is constant.
- Load is increased => speed tends to decrease and vice versa.
- The function of the governor is to regulate the quantity of water flowing through the runner in proportion to the load. Thus the governing mechanism maintains the speed of the runner at a constant level at all loads.
- For reaction turbines, the governor controls the guide vanes and wicket gates. For impulse turbines, the governor controls the spear and nozzle.

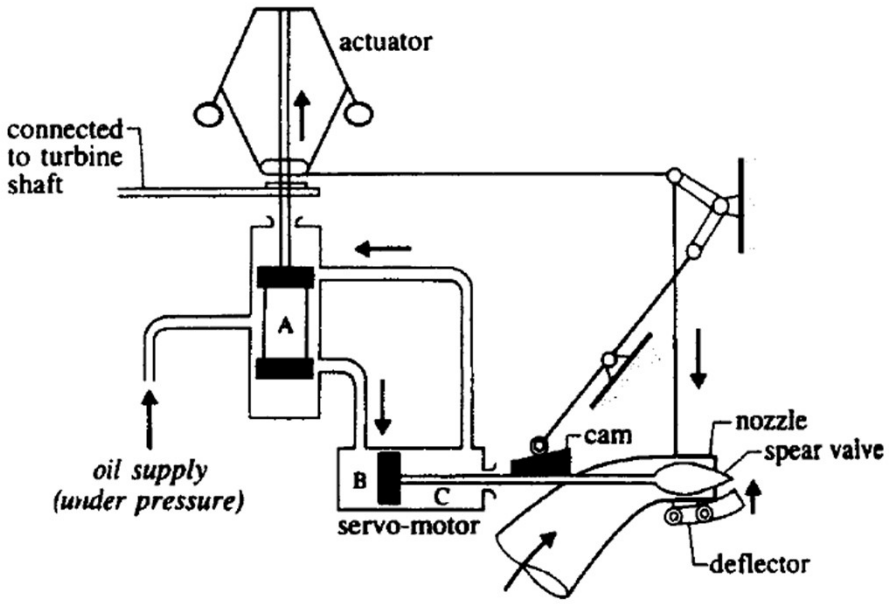
Governor

The governor is a mechanism controlling the rotational speed of the turbo-generator unit; constant speed must be maintained in order to obtain the a.c. supply with constant frequency. As the turbine and hence its interconnected generator tend to decrease or increase speed as the load varies, the maintenance of an almost constant speed requires regulation of the amount of water allowed to flow through the turbine by closing or opening the gates (or nozzles) of the turbine automatically, through the action of a governor.

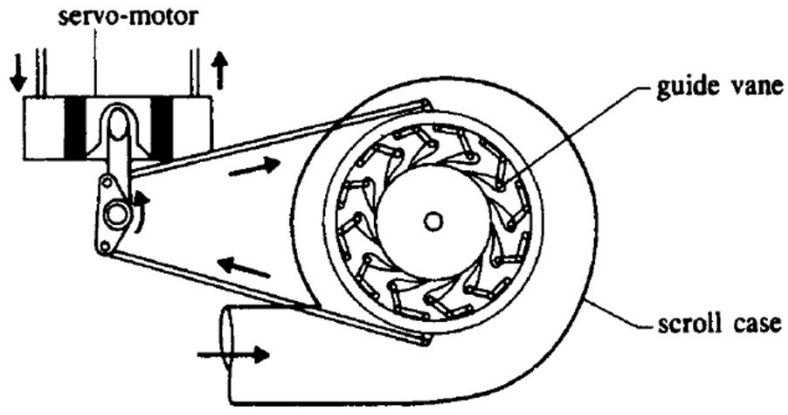
A simple governing mechanism for turbines is shown in figure below. Increase in the rotor speed raises piston A, permitting oil to enter chamber B, thus closing the gates slightly. The operation is reversed if the speed drops.

A rapid closing or opening of the nozzle or guide vanes (gates) is undesirable, as serious water hammer problems may result in the penstocks.

(a) Governing of impulse turbine



(b) Governing of reaction turbine



Sudden changes may be avoided in the case of a Pelton turbine if a deflector is activated in front of the jet, thus diverting part of the flow away from the turbine. Similarly, in the case of a reaction-type turbine a relief valve may allow a part of the discharge to flow directly to the tail race without entering the runner.

17.12 Governor and its mechanism

The governing of a turbine is defined as the operation by which the speed of turbine is kept constant under all working condition. It is done automatically by means of governer, which regulates the rate of flow through turbine, according to the changing load condition of the turbine. Governing of the turbine means regulating the speed of the turbine.

Governing of a turbine is necessary as a turbine is directly coupled to an electric generator, which is required to run at constant speed under fluctuation load condition. The frequency of power generation by a generator of a constant number of pair of poles under varying condition should be same.

When the load on the generator decreases the speed of the generator increases beyond the normal speed. If the turbine or the generator is to run at constant speed, the rate of flow of water to the turbine should be decreased till the speed becomes normal. The process by which the speed of the turbine is kept constant under varying condition of load is called governing.

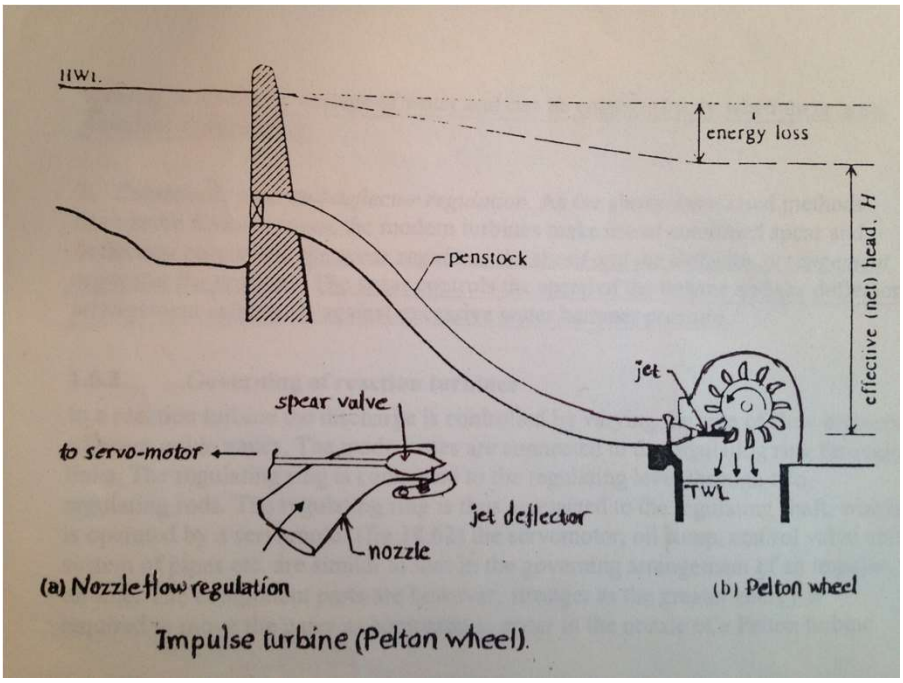
The governor of a Pelton wheel turbine decreases or increases the outlet area of the nozzle by moving the needle valve. In the case of Francis or Kaplan turbine, the governor decreases or increases the wicket gate.

The function of the governor is to detect any error in speed between actual and desired values and to effect a change in the turbine output. This is done so that the system load is in equilibrium with the generating unit output at the desired speed. The governor system of the turbine acts as opening, closing and gate-setting system for starting, stopping and synchronizing the turbine which allows for matching output to the system load to maintain the system frequency.

1.6.1 Governing of impulse turbines

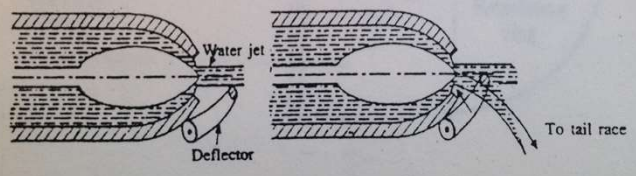
In order to regulate the quantity of water rejected from the turbine nozzle and from striking the buckets one of the following methods of regulation may be adapted:

1.Spear regulation. 2.Deflector regulation. 3.Combined spear and deflector regulation



Impulse turbine (Pelton wheel).

1. *Spear regulation.* In this method the rate of flow is regulated by altering the cross-sectional area of stream by moving the spear to and fro inside nozzle. This method of speed regulation is suitable when the fluctuation of load is small and a relatively large penstock feeds a small turbine. The disadvantage of this method is that when a load falls all of sudden, the turbine nozzle has to close suddenly which may cause water ~~haunter~~ hammer in the penstock.



(*a*) Full load (*b*) Deflector diverts part of the jet

Deflector regulation.

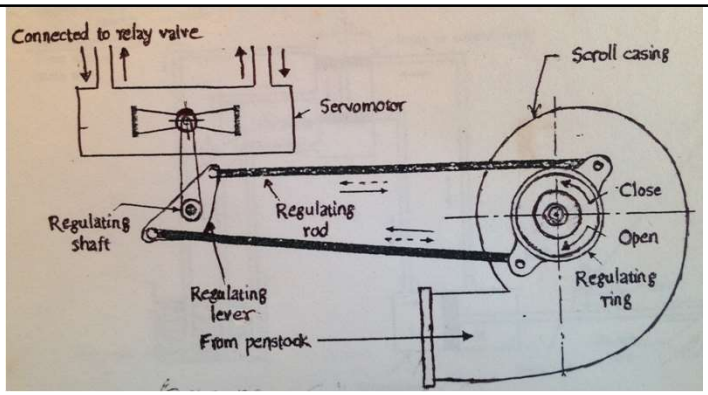
2. *Deflector regulation.* The deflector is generally a plate connected to the oil pressure governor by means of levers. When necessity arises to deflect the jet the plate can be brought in between the nozzle and buckets, thereby diverting the water away from the runner and directing into the tailrace. The use of deflector regulation is restored to when the supply of water is constant but the load fluctuates. The position of spear can be adjusted by hand. As the nozzle has always a constant

opening, it results in wastage of water and can be employed only when there is an abundant water supply.

3. *Combined spear and deflector regulation.* As the above-mentioned methods have some disadvantages, the modern turbines make use of combined spear and deflection regulation; *the spear regulates the speed and the deflector arrangement regulates the pressure*. The spear controls the speed of the turbine and the deflector arrangement safeguards against excessive water hammer pressure.

1.6.2 Governing of reaction turbines

In a reaction turbine the discharge is controlled by varying the area of flow between adjacent guide vanes, The guide vanes are connected to the regulating ring through links. The regulating ring is connected to the regulating lever through two regulating rods. The regulating ring is thus connected to the regulating shaft, which is operated by a servomotor (fig.18.62) the servomotor; oil pump, control valve and system of pipes etc. are similar to that in the governing arrangement of an impulse turbine. The component parts are however, stronger as the greater energy is required to move the gates as compared to spear in the nozzle of a Pelton turbine.



1.6.3 Governor Controlling Systems

Mainly Governor system can be classified as:

- Mechanical Hydraulic Governor
- Electro Hydraulic Governor

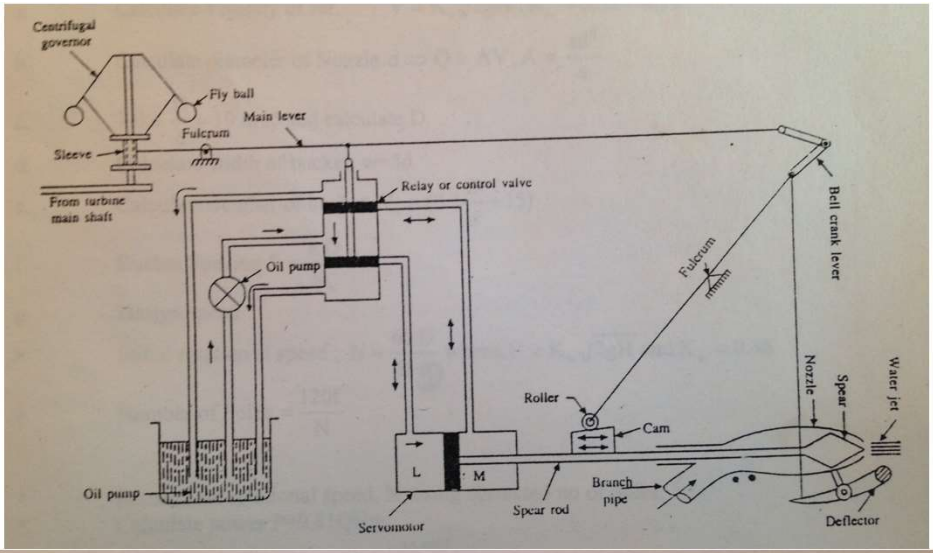
In any type of governor the basic three elements of operation are:

- The speed responsive system for detecting changes in speed

- The power component for operating wicket gates/spear valve etc.
- The stabilizing element that prevents runaway speed in the turbine and holds the servomotor in position which the turbine output and generator load are equalized.

Mechanical Hydraulic Governor : The working of the system is as follows:

- When the load on the turbine increases the speed of runner falls and consequently balls of the centrifugal governor move inward; the governor sleeve moves downward.
- The downward movement of the sleeve is transmitted to a relay or control valve (through suitable linkages), which admits oil under pressure to a servomotor. The oil exerts a force on the piston of servomotor, and that puss the spear to a position that increases the annular area of the nozzle flow passage; the quantity of water striking the buckets is then increased and the turbine regains its normal speed.



Electric Governing : The electric governor uses the electronic circuits in place of mechanical devices.

Governing of Turbines

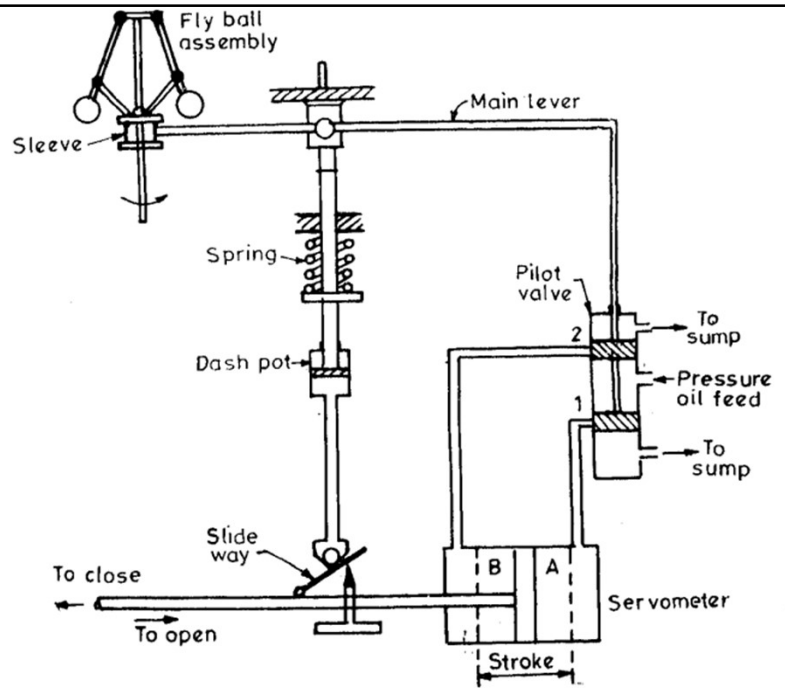
Governing of turbine means speed regulation. The turbine governor is an essential piece of equipment of the unit and works to regulate the speed of the turbine upon changes in load, by controlling the flow of water through the runner.

The various components of the governor are :

(*i*) a speed sensitive device (fly ball assembly), (*ii*) a mechanical amplifier to increase the effect of the above device, and (*iii*) the follow up system which produces the actual control on the water flow. This actual control comprises of operating a gate mechanism in the case of Francis turbine, the gate and blade adjusting mechanism in the case of Kaplan turbine, and the nozzle, needles and deflectors in an impulse turbine (Pelton wheel).

The most commonly used system of governing in turbines is through oil pressure. The oil pressure governor is described here.

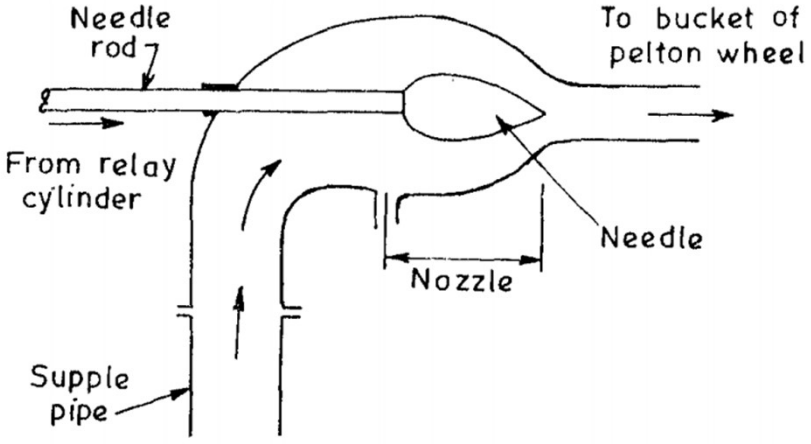
The governor system includes the governer itself, the energy supply equipment (pump supplying oil under constant pressure), the piping and the connections to the turbine's flow control mechanism.



Turbine flow control mechanism.

The governor in its simple form consists of the governor head (speed sensitive device, usually flyball assembly), the control or pilot valve and the servomotor. At normal speed the ports 1 and 2 of the control valve are both closed and the pressure in chambers *A* and *B* of the servomotor being the same, the servomotor piston is stationary in position, maintaining the constant rated discharge. Increase in the turbine speed rotates (through a belt) the governor head faster. The flyballs move outwards lifting the sleeve up and opening the port 1. Oil pressure is applied in chamber *A*, which moves the piston to the left and reduces the discharge until normal speed is restored. Any decrease in the turbine speed brings the sleeve downwards, opening the port 2, thereby applying pressure in chamber *B* and pushing the piston to the right to increase the discharge until the speed is stepped up to the normal. This apparently is simple process, however is rendered unsatisfactory because of the inertia effects, which cause the over travel of the controls and the consequent 'hunting' of governor.

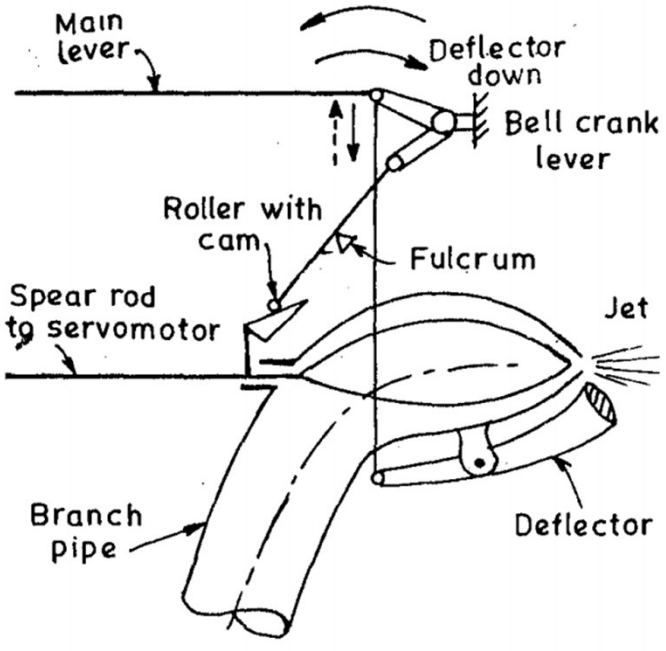
As has been stated, the governing of an impulse turbine is effected by regulating the quantity of water ejected through the nozzles and striking the turbine buckets. Two common ways of doing this are : Spear or needle regulation and the deflector regulation A third method involves the combination of these two arrangements. Fig. below shows the guide mechanism for an impulse turbine



Guide mechanism for an impulse turbine.

using needle regulation. The governor closes the nozzle opening partially by pushing the needle in the nozzle thus allowing less water to enter the turbine, which, therefore, will run slow. Needle regulation is suitable where a relatively large penstock supplies water to a small turbine and where the fluctuations in load are small, while the deflector is generally employed where supply of water is constant but the load on the turbine fluctuates.

The combination of spear and deflector control is quite popular in modern installations since double regulation is a feature of all modern turbines. The double operation involves simultaneous operation of two elements. This uses both the needle regulation effected through the direct connection with governor servomotor piston rod, and the deflector regulation which is effected through a linkage arrangement from the main lever and the servomotor piston rod.. The deflector is thus, actuated directly by the governor and works quicker than the needle/ spear regulator which is permitted to act gradually, and thus avoid serious water hammer in the piping system. The double regulation arrangement of turbine governing is shown in Fig. below .



Double regulation system of governing impulse turbine

Double regulation system is also employed in reaction turbines by control the water flow through guide blades. This consists of guide blade control and a relief valve in Francis turbine and guide blade control and runner blade adjustment in case of Kaplan turbine. The guide blade control comprises of the used control mechanism, and blade adjusting mechanism are additional features. Relief valve which fitted in the penstock at the turbine inlet, opens and direct the water to the tailrace by passing the runner, when the power demand drops suddenly and the guide blades are closed suddenly.

Electro-mechanical Installation

Generator:

Electric Generator is a machine that converts mechanical energy to electric energy. Usually this energy is obtained from a rotating shaft that is also called the armature of the generator. The electric energy then produced can be used for power transmission to commercial, industrial or even domestic level. Generators supply current which usually has a frequency of 50 Hz, which is used here. An electric generator has two parts: i)Stator and ii) Rotor

The stator comprises of the stationary magnetic poles, whereas the rotating armature is included in the rotor.

Hydropower generator:

Hydropower generator is the main power equipment to making electricity energy in a hydropower station. It is a water-to-wire generator with the turbine as the prime motor. When the water flow goes through turbine, it changes the water power into mechanical energy.

Electro-mechanical Installation

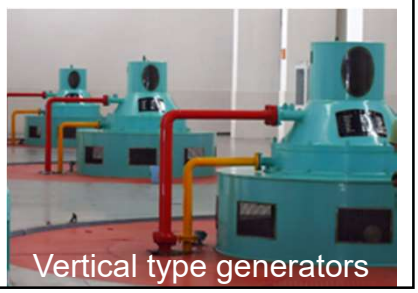
Types of hydropower generator:

According to its axis location, hydro generator is usually divided into two types: i) Horizontal and ii) Vertical

Large and medium-sized units usually adopt the vertical type layout, whose maximum speed can reach 750 rpm. While horizontal type usually is used for medium and small capacity units, whose maximum speed can reach 1500 rpm.

Usually, the horizontal hydro generator drives by Pelton turbine, and the vertical hydro generator drives by Francis or Kaplan turbines.

Horizontal type generators



Vertical type generators

Generators transform mechanical energy into electrical energy. Although most of early hydroelectric systems were of direct current variety to match early commercial electrical system now a days only 3-phase alternating current generators are used in normal practice. Depending upon the practice of network supply, there are 2 types of generators

a) Synchronous Generator: Such generators are equipped with a DC excitation systeem (rotating or static) associated with a voltage regulator to provide voltage and phase angle control before the generators is connected to the grid. Synchronous generators can run isolated from the grid and produce power since excitation is not grid dependent. These are more expensive than asynchronous generator.

b) Asynchronous Generator: These are simple squirrel cage induction motors with no possibility of voltage regulation and running at a speed directly related to system frequency. They draw their excitation current from the grid, absorbing reactive energy by their own magnetism. They can not generate when disconnected from the grid because they ar incapable of providing their own excitation current.

Rating of Generators:

Generator rating is a set of specified values which are specified for that particular generator model by its manufacturers. It must be capable of supplying that much power output in an accurate way that is anticipated by the manufacturer.

The sizes of the generators vary depending on their rating and on their shaft arrangements (either vertical or horizontal). A.C. generators are rated in kilovolt-amperes (kVA). The apparent or nominal rating (output) differs from the actual output, Pa, the difference Pm being used to magnetize the rotor field. Thus the rated kVA is given by

$$\text{kVA} = (P_a^2 + P_m^2)^{1/2}$$

and the power factor is given by

$$PF = \cos\phi = P_a/\text{kVA}$$

Pump:

A pump is a device that moves fluids (liquids or gases), or sometimes slurries, by mechanical action. Pumps operate by some mechanism (typically reciprocating or rotary), and consume energy to perform mechanical work by moving the fluid.

19·2. Classification of pumps

On the basis of transfer of mechanical energy the pumps can be broadly *classified* as follows

1. *Rotodynamic pumps*

(*i*) Radial flow pumps (*ii*) Axial flow pumps (*iii*) Mixed flow pumps.

2. *Positive displacement pumps.*

In rotodynamic pumps, *increase in energy level is due to a combination of centrifugal energy, pressure energy, and kinetic energy.*

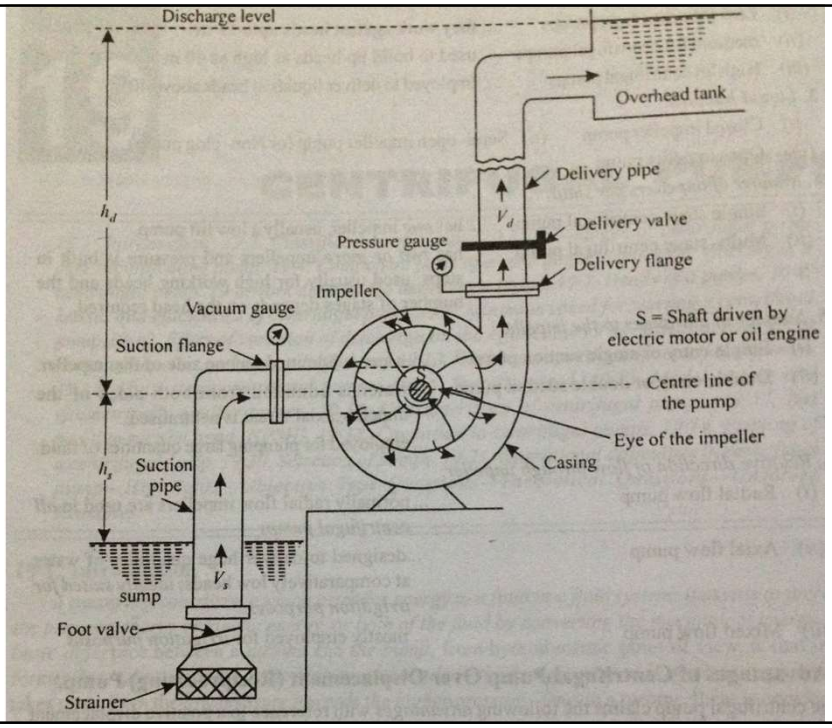
The radial flow type pumps are commonly called *centrifugal pumps* (only these pumps will be discussed in this chapter).

19·4. Component Parts of a Centrifugal Pump

Refer. Fig. 19·1. A centrifugal pump consists of the following *main components*:

1. Impeller 2. Casing 3. Suction pipe 4. Delivery pipe.

1. **Impeller.** An *impeller* is a wheel (or rotor) with a series of backward curved vanes (or blades). It is mounted on a shaft which is usually coupled to an electric motor.



2. **Casing.** The casing is an *airtight chamber surrounding the pump impeller.* It contains suction and discharge arrangements, supporting for bearings, and facilitates to house the rotor assembly .It has provision to fix stuffing box and house packing materials which prevent external leakage. The essential *purposes* of the casing are:

(*i*) To guide water to and from the impeller, and

(*ii*) To partially convert the kinetic energy into pressure energy.

3. **Suction pipe.** The pipe which connects the centre/eye of the impeller to sump from which liquid is to be lifted is known as *suction pipe.* In order to check the formation of air pockets the pipe is laid *air tight.* To prevent the entry of soid particles, debris etc. into the pump the suction pipe is provided with a strainer at its lower end.The lower end of the pipe is also fitted with a *non - return foot valve* which does not permit the liquid to drain out of the suction pipe when pump is *not working*; this also helps in *priming.*

4. **Delivery pipe.** The pipe which is connected at its lower end to the outlet of the pump and it delivers the liquid to the required height is known as *delivery pipe* . A *regulating valve* is provided on the delivery pipe to regulate the supply of water.

19·5. Working of a Centrifugal pump

A centrifugal pump works on the *principal that when a certain mass of fluid is rotated by an external source, it is thrown away from the central axis of rotation and a centrifugal head is impressed which enables it to rise to a higher level.*

The *working /operation* of a centrifugal pump is explained *step-wise* below:

1. The delivery valve is closed and the pump is *primed* that is, *suction pipe, casing and portion of the delivery pipe upto the delivery valve are completely filled with the liquid (to be pumped) so that no air pocket is left.*

2. Keeping the delivery valve still closed the electric motor is started to rotate the impeller. The rotation of the impeller causes strong suction or vaccum just at the eye of the casing.

3. The speed of the impeller is gradually increased till the impeller rotates at its normal speed and develops normal energy required for pumping the liquid.

4. After the impeller attains the normal speed the delivery valve is opened when the liquid is continuously sucked (from sump well) up the suction pipe , it passes through the eye of casing and enters the impeller at its centre or it enters the impeller vanes at their inlet tips. This liquid is impelled out by the rotating vanes and it comes out at the outlet tips of the vanes into the casing. Due to impeller action the *pressure head as well as velocity heads of the liquid are increased* (some of this velocity heads is converted into pressure head in the casing and in the diffuser blades/vanes if they are also provided).

5. From casing , the liquid passes into pipe and is lifted to the required height (and discharged from the outlet or upper end of the delivery pipe).

6. So long as motion is given to the impeller and there is supply of liquid to be lifted the process of lifting the liquid to the required height remains continuous.

7. When pump is to be stopped the delivery valve should be first closed, otherwise there may be some backflow from the reservoir.

19·16. Characteristics of Centrifugal Pumps

Ordinarily a centrifugal pump is worked under its maximum efficiency conditions. However, when the pump is run at conditions different from the design conditions, it performs differently. Therefore, to predict the behaviour of the pump under varying conditions of speeds,heads, discharges or powers, tests are usually conducted. The results obtained from these tests are plotted in from of *characteristic curves;* these curves delineate useful information about the performance of a pump in its installation.

The following *four types of characteristic curves* are usually prepared for centrifugal pumps:

1. Main characteristic curves,
2. Operating characteristic curves,
3. Constant efficiency or Muschel curves, and
4. Costant head and constant discharge curves.

1. Main characteristic curves:

The main characteristic curves are obtained as follows:

- The pump is run at a *constant speed* and the *discharge is varied* over the desired range (by delivery valve)
- Measurements are taken for manometric head (H_{mano}) and shaft power (P) for each discharge(Q).
- Calculations are made for the pump overall efficiency, η_0
- The curves are plotted between Q and H_{mano}; Q and P; and Q and η_0 for that speed.
- The same procedure is repeated by running the pump at another speed.
- A family of curves is obtained as shown in Fig. 19·25.

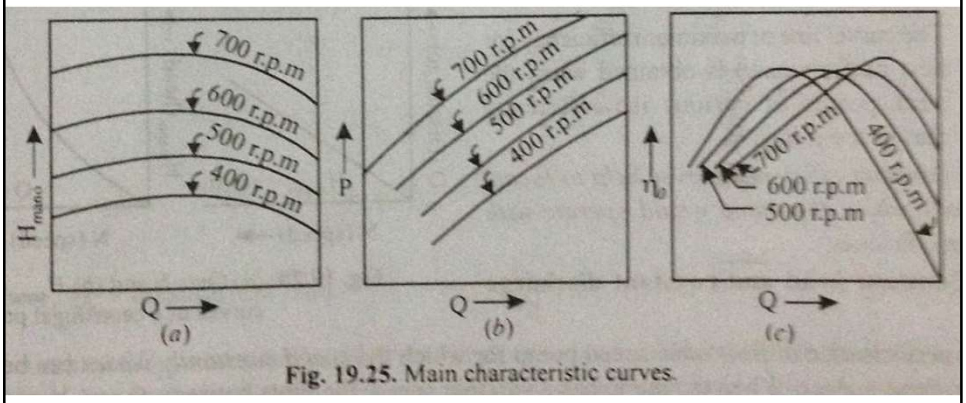


Fig. 19.25. Main characteristic curves.

2. Operating characteristic curves:

When a centrifugal pump operates at the *design speed* (same as speed of driving motor) the *maximum* efficiency occurs. Evidently for *optimum performance,* the pump needs to be operated at the design speed. To obtain *operating characteristic curves* the pump is run at the design speed and the discharge is varied, as in the case of main characteristic curves. The operating characteristic curves are shown in Fig. 19·26 .The design discharge and head *are obtained from the corresponding* curves where the efficiency is maximum,

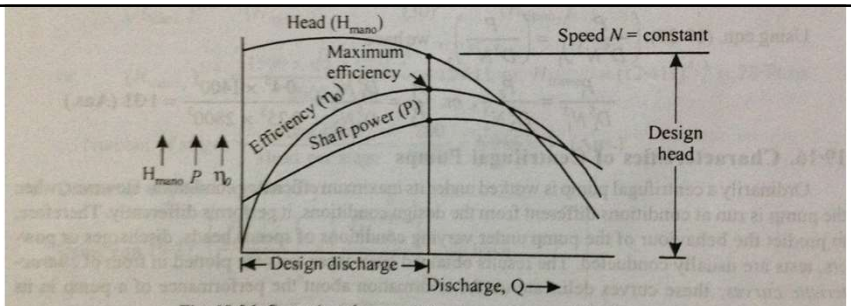


Fig. 19.26. Operating characteristic curves of a centrifugal pump.

3. Constant efficiency or Muschel curves:

The constant efficiency curves (also called iso-efficiency curves), depict the performance of a pump over its entire range of operations. These curves are obtained from main characteristic curves as follows:

- For a given efficiency, the values of discharges are obtained from Fig. (19·25) (c). These points are projected on the head (H_{mano}) *v/s* discharge (Q) for that speed in Fig. 19·25 (a).
- Similarly, for another value of efficiency and speed, the points are obtained and projected.
- The points corresponding to one efficiency are joined.

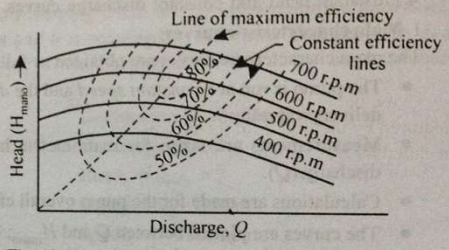


Fig. 19.27. Constant efficiency or Muschel curves.

- The curves so obtained are the constant efficiency or Muschel curves.
- The curve/ line of maximum efficiency (or best performance) is obtained when the peak points of various iso-efficiency curves are joined.

The constant efficiency curves help to locate the regions where the pump would operate with maximum efficiency.

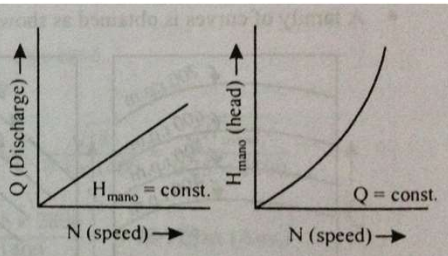


Fig. 19.28. (*a*) Qv/s *N* and (*b*) H_{mano} v/s *N* curves of a centrifugal pump.

4. **Constant head and constant discharge curves:**

The performance of a variable speed pump for which the *speed constantly varies* can be determined by these curves. When the pump has a variable speed, the plots between *Q* and *N*, and H_{mano} and *N* may be obtained. In the first case H_{mano} is kept constant and in the second case, *Q* is kept constant. The curves are shown in Fig. 19·28.

Reciprocating Pump

The reciprocating pump is a *positive displacement pump* as it sucks and raises the liquid by actually displacing it with a piston/plunger that executes a reciprocating motion in a closely fitting cylinder. The amount of liquid pumped is equal to the volume displaced by the piston.

The pumps designed with disk pistons create pressures upto 25 bar and the plunger pumps built up still higher pressures. Discharge from these pumps is almost wholly dependent on the pump speed.

The total efficiency of a reciprocating pump is about 10 to 20% higher than a comparable centrifugal pump.

Reciprocating pumps for industrial uses have almost become *obsolete owing to their high capital cost as well as maintenance cost as compared to that of centrifugal pumps*. However, small hand-operated pumps such as cycle pumps, football pumps, kerosene pumps, village well pumps and pumps used as important parts of hydraulic jack etc. still find wide applications. *The reciprocating pump is best suited for relatively small capacities and high heads*. This type of pump is very common in *oil drilling operations*.

20·3. Main Components and Working of a Reciprocating Pump

Refer Fig. 20·1. The **main parts** *of a reciprocating pump are*:

1. Cylinder
2. Piston
3. Suction valve
4. Delivery valve
5. Suction pipe
6. Delivery pipe
7. Crank and connecting rod mechanism operated by a power source e.g. steam engine, internal combustion engine or an electric motor.

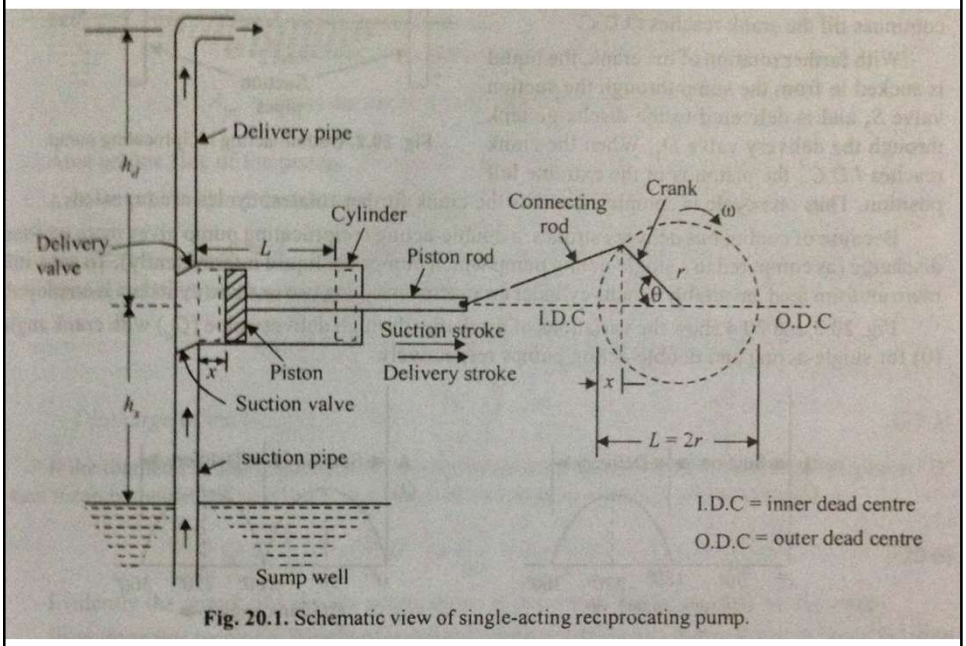


Fig. 20.1. Schematic view of single-acting reciprocating pump.

Working of a single-acting reciprocating pump:

As shown in Fig. 20·1 a single acting reciprocating pump has one suction pipe and one delivery pipe. It is usually placed above the liquid level in the sump. When the crank rotates the piston moves backward and forward inside the cylinder. The pump operates as follows:

— Let us suppose that initially the crank is at the inner dead centre (I.D.C.) and crank rotates in the clockwise direction. As the crank rotates, the piston moves towards right and a vacuum is created on the left side of the piston. This vacuum causes suction valve to open and consequently the liquid is forced from the sump into the left side of the pistor When the crank is at the outer dead centre (O.D.C) the suction stroke is completed and the left side of the cylinder is full of liquid.

When the crank further turns from O.D.C to I.D.C., the piston moves inward to the left and high pressure is built up in the cylinder. The delivery valve opens and the liquid is forced into the delivery pipe. The liquid is carried to the discharge tank through the delivery pipe. At the end of delivery stroke the crank comes to the I.D.C and the piston is at the extreme left position.

Working of a double-acting reciprocating pump:

Refer Fig. 20·2. In a double-acting reciprocating pump, suction and delivery *strokes occur simultaneously*. When the crank rotates from *I.D.C.* in the clockwise direction, a vaccum is created on the left side of piston and the liquid is sucked in from the sump through value S_1. At the same time, the liquid on the right side of the piston is pressed and a high pressure causes the delivery valve D_2 to open and the liquid is passed on to the discharge tank. This operation continues till the crank reaches *O.D.C.*

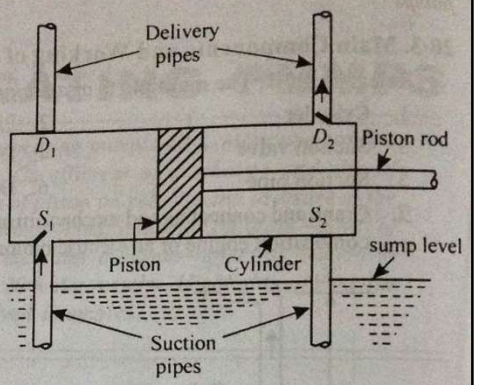
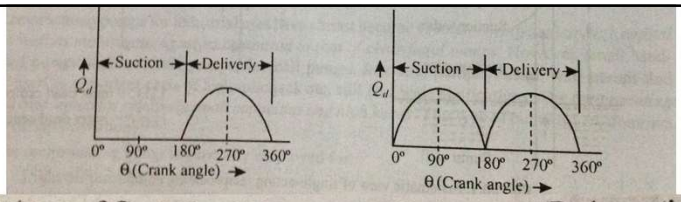


Fig. 20.2. Double-acting reciprocating pump.

With further rotation of the crank, the liquid is sucked in from the sump through the suction valve S_2 and is delivered to the discharge tank through the delivery valve D_1. When the crank reaches *I.D.C.*, the piston is in the extreme left position. Thus one cycle is completed and as the crank further rotates, cycles are repeated.

Because of continuous delivery strokes, a double-acting reciprocating pump gives more uniform discharge (as compared to a single-acting pump which pumps the liquid intermittently). To get a still more uniform feed, invariably a multi-cylinder arrangement having two or more cylinders is employed.

Fig. 20·3 and 20·4 show the variations of discharge through delivery pipe (Q_d) with crank angle (θ) for single-acting and double-acting pumps respectively.



19·3. Advantages of Centrifugal Pump Over Displacement (Rciprocating) Pump.

The centrifugal pump claims the following *advantages* with reference to a positive displacement (reciprocating) pump.

1. The cost of a centrifugal pump is less as it has fewer parts.
2. Installation and maintenance are easier and cheaper.
3. Its discharging capacity is much greater than that of a reciprocating pump.
4. It is compact and hads smaller size and weight for the same capacity and energy transfer.
5. Its performance characteristics are superior.
6. It can be employed for lifting highly viscous liquid such as paper pulp, muddy and sewage water ,oil, sugar molasses etc.
7. It can be operated at very high speeds without any danger of separation and cavitation.
8. It can be directly coupled to an electric motor or an oil engine.
9. The torque on the power source is uniform,the output fron the pump is also uniform.

However , because of higher efficiency the reciprocating pumps are still employed for high heads and small discharges. A reciprocating pump can build up very high pressures (as high as 700 bar or even more) and as such these pumps are used for lifting oils from very deep oil wells.

Example 18·11. *The following data relate to a Pelton wheel :*

Head	*... 72 m*
Speed of the wheel	*... 240 r.p.m.*
Shaft power of the wheel	*...115 kW*
Speed ratio	*... 0·45*
Co-efficient of velocity	*... 0·98*
Overall efficiency	*... 85%*

Design the Pelton wheel.

Solution. Effective head, $H = 72$ m

Speed of the wheel, $N = 240$ r.p.m.

Shaft power, $P = 115$ kW

Speed ratio, $K_u = 0{\cdot}45$

Co-efficient of velocity, $C_v = 0{\cdot}98$

Overall efficiency, $\eta_0 = 85\%$

Design of the Pelton wheel means to find diameter of the wheel D, diameter of jet (d), width and depth of buckets and number of buckets on the wheel.

(*i*) **Diameter of wheel, D:**

Velocity of jet, $V_1 = C_v\sqrt{2gH} = 0{\cdot}98\sqrt{2 \times 9{\cdot}81 \times 72} = 36{\cdot}8$ m/s

$\therefore$ Bucket velocity, $u\,(= u_1 = u_2) = K_u \times V_1 = 0{\cdot}45 \times 36{\cdot}8 = 16{\cdot}56$ m/s

But, $u = \dfrac{\pi DN}{60}$, or, $D = \dfrac{60u}{\pi N} = \dfrac{60 \times 16{\cdot}56}{\pi \times 240} = \mathbf{1{\cdot}32\ m}$ **(Ans.)**

(*ii*) **Diameter of jet, d:**

Overall efficiency, $\eta_0 = \dfrac{\text{Shaft power}}{\text{Water power}} = \dfrac{P}{wQH}$

$$0.85 = \frac{115}{9.81 \times Q \times 72}$$

or, $$Q = \frac{115}{0.85 \times 9.81 \times 72} = 0.1915\ \text{m}^3/\text{s}$$

But, $Q =$ Area of jet $\times$ velocity of jet

$$0.1915 = \frac{\pi}{4} \times d^2 \times V_1 = \frac{\pi}{4} d^2 \times 36.8$$

$\therefore$ $$d = \left(\frac{0.1915 \times 4}{\pi \times 36.8}\right)^{1/2} = 0.0814\ \text{m or } \mathbf{81.4\ mm}\ \textbf{(Ans.)}$$

(*iii*) **Size of buckets:**

Width of the bucket, $B = 3$ to 4 times jet diameter (d)

$\simeq 3{\cdot}5\,d = 3{\cdot}5 \times 81{\cdot}4 = \mathbf{285\ mm}$ **(Ans.)**

Radial length of bucket, $L = 2$ to 3 times jet diameter (d)

$\simeq 2{\cdot}5\,d = 2{\cdot}5 \times 81{\cdot}4 = \mathbf{203{\cdot}5\ mm}$ **(Ans.)**

Depth of bucket, $T = 0{\cdot}8$ to $1{\cdot}2$ times jet diameter (d)

$\simeq 1{\cdot}0\,d = \mathbf{81{\cdot}4\ mm}$ **(Ans.)**

(*iv*) **Number of buckets on the wheel, Z :**

$$Z = 15 + \frac{D}{2d} = 15 + \frac{1{\cdot}32 \times 1000}{2 \times 81{\cdot}4} = \mathbf{23}\ \textbf{(Ans.)}$$

Example 18·43. *A conical draft tube having inlet and outlet diameters 1·2 m and 1·8 m discharges water at outlet with a velocity of 3 m/s. The total length of the draft tube is 7·2 m and 1·44 m of the length of draft tube is immersed in water. If the atmospheric pressure head is 10·3 m of water and loss of head due to friction in the draft tube is equal to 0·2 × velocity head at outlet of the tube, determine :*

(*i*) *Pressure head at inlet, and*

(*ii*) *Efficiency of the draft tube.*

Solution. Inlet diameter of the draft tube, $d_i = 1{\cdot}2$ m

Outlet diameter, $d_0 = 1{\cdot}8$ m

Velocity at outlet, $V_3 = 3$ m/s

Total length of draft tube, $H_s + y = 7{\cdot}2$ m

Length of draft tube in water, $y = 1{\cdot}44$ m

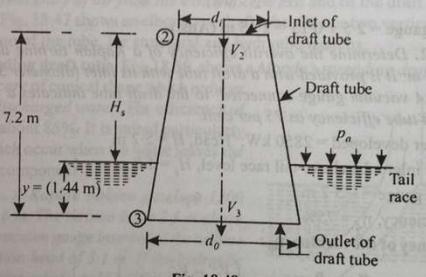


Fig. 18.49

$\therefore \quad H_s = 7{\cdot}2 - 1{\cdot}44 = 5{\cdot}76$ m

Atmospheric pressure head, $\frac{p_a}{w} = 10{\cdot}3$ m

Loss of head due to friction,

$h_f = 0{\cdot}2 \times$ velocity head at outlet

$$= 0{\cdot}2 \frac{V_3^2}{2g}$$

(*i*) **Pressure head at inlet, $\frac{p_2}{w}$:**

Discharge through the draft tube,

$$Q = A_3 V_3 = \frac{\pi}{4} \times d_0^2 \times V_3 = \frac{\pi}{4} \times 1{\cdot}8^2 \times 3 = 7{\cdot}634 \text{ m/s}$$

Velocity of inlet, $V_2 = \frac{Q}{A_2} = \frac{7{\cdot}634}{\frac{\pi}{4} d_i^2} = \frac{7{\cdot}634}{\frac{\pi}{4} \times 1{\cdot}2^2} = 6{\cdot}75$ m/s

Using eqn. (18·32),

$$\frac{p_2}{w} = \frac{p_a}{w} - H_s - \left(\frac{V_2^2 - V_3^2}{2g} - h_f \right) = \frac{p_a}{w} - H_s - \left(\frac{V_2^2 - V_3^2}{2g} - 0{\cdot}2 \frac{V_3^2}{2g} \right)$$

$$= 10{\cdot}3 - 5{\cdot}76 \left(\frac{6{\cdot}75^2 - 3^2}{2 \times 9{\cdot}81} - 0{\cdot}2 \times \frac{3^2}{2 \times 9{\cdot}81} \right)$$

or, $\frac{p_2}{w} = 4{\cdot}54 - (1{\cdot}863 - 0{\cdot}092) = \mathbf{2{\cdot}769}$ **m (abs) (Ans.)**

(*ii*) **Efficiency of the draft tube, η_d :**

$$\eta_d = \frac{\left(\frac{V_2^2 - V_3^2}{2g} - h_f \right)}{\frac{V_2^2}{2g}} = \frac{\frac{V_2^2 - V_3^2}{2g} - 0{\cdot}2 \frac{V_3^2}{2g}}{\frac{V_2^2}{2g}} = \frac{\frac{V_2^2}{2g} - \left(\frac{V_3^2}{2g} + 0{\cdot}2 \frac{V_3^2}{2g} \right)}{\frac{V_2^2}{2g}}$$

$$= 1 - 1{\cdot}2 \left(\frac{V_3}{V_2} \right)^2 = 1 - 1{\cdot}2 \left(\frac{3}{6{\cdot}75} \right)^2 = 0{\cdot}763 \quad \text{or} \quad \mathbf{76{\cdot}3\ \%}\ \textbf{(Ans.)}$$

Example 18·45. *A turbine is to operate under a head of 25 m at 200 r.p.m . The discharge is 9 m³/s. If the overall efficiency is 90 per cent, determine :*

(*i*) *Power generated;* (*ii*) *Specific speed of the turbine;*

(*iii*) *Type of turbine.* **[AMIE]**

Solution. Head, $H = 25$ m; Speed, $N = 200$ r.p.m.;

Discharge, $Q = 9$ m³/s; Overall efficiency, $\eta_0 = 90\%$.

(*i*) **Power generated, P :**

$$P = \eta_0 \times wQH = 0{\cdot}9 \times 9{\cdot}81 \times 9 \times 25 = \mathbf{1986{\cdot}5\ kW\ (Ans.)}$$

(*ii*) **Specific speed of the turbine, N_s :**

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{200 \times \sqrt{1986{\cdot}5}}{(25)^{5/4}} = \mathbf{159{\cdot}4\ r.p.m.\ (Ans.)}$$

(*iii*) **Type of Turbine :**

As the specific speed lies between 80 and 400 (Refer to table 18·2), the turbine is a **Francis turbine. (Ans.)**

Example 18·48. *In a hydroelectric station, water is available at the rate of 175 m³/s under a head of 18 m. The turbines run at a speed of 150 r.p.m. with overall efficiency of 82%. Find the number of turbines required if they have the maximum specific speed of 460.* **(GATE)**

Solution. *Given* : $Q = 175$ m³/s; $H = 18$ m; $N = 150$ r.p.m.; $\eta_0 = 82\%$; $N_s = 460$.

Number of turbines required :

Specific speed of the turbine,

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} \qquad \text{...[Eqn. (18.33)]}$$

$$460 = \frac{150\sqrt{P}}{(18)^{5/4}} \quad \text{(where, P is in kW and H is in metres.)}$$

or, Power available at turbine shaft, $P = \left[\frac{460 \times (18)^{5/4}}{150}\right]^2 = 12927{\cdot}5$ kW

Power available from turbines $= wQH \times \eta_0 = 9.81 \times 175 \times 18 \times 0.82 = 25339.23$ kW

No. of turbines required $= \frac{25339{\cdot}23}{12927{\cdot}5} = 1{\cdot}96$ say **2 (Ans.)**

Example 18·58. *A Francis turbine works under a head of 25 m and produces 11800 kW while running at 120 r.p.m. The turbine has been installed at a station where atmospheric pressure is 10 m of water and vapour pressure is 0·2 m of water. Calculate the maximum height of the straight draft tube for the turbine.*

Solution. Head under which the turbine works, $H = 25$ m

Power output, $P = 11800$ kW

Speed of the turbine, $N = 120$ r.p.m.

Atmospheric pressure, $p_a = 10$ m of water

Vapour pressure, $H_v = 0{\cdot}2$ m.

Maximum height of the draft tube, H_s:

Specific speed, $N_s = \dfrac{N\sqrt{P}}{H^{5/4}} = \dfrac{120\sqrt{11800}}{(25)^{5/4}} = 233{\cdot}2$ r.p.m.

Critical value of Thoma's cavitation factor for a Francis turbine

$$\sigma_c = 0{\cdot}625\left(\frac{N_s}{380{\cdot}78}\right)^2 \quad \text{...[Eqn. (18·49)]}$$

$$= 0{\cdot}625\left(\frac{233{\cdot}2}{380{\cdot}78}\right)^2 = 0{\cdot}2344$$

Also, $\sigma_c = \dfrac{H_c - H_v - H_s}{H}$ ...By definition

or, $0{\cdot}2344 = \dfrac{10 - 0{\cdot}2 - H_s}{25}$, or, $0{\cdot}2344 \times 25 = 10 - 0{\cdot}2 - H_s$

$\therefore \quad H_s = 10 - 0{\cdot}2 - 0{\cdot}2344 \times 25 = 3{\cdot}94$ m

Hence, *maximum permissible height of the draft tube* = **3·94 m (Ans.)**

Solved Examples 17-1

A conical draft tube having inlet and output diameters 1.2 and 1.8 m discharges water at output with a velocity of 3 m /sec. The total length of the draft tube is 7.2 m and 1.44 m of length of draft tube is immersed in water. If atmospheric pressure head is 10.3 m of water and loss of head due to friction in the draft tube is equal to 0.2 times velocity head at output of the tube. Determine a) pressure head at input and b) Efficiency of draft tube.

Solution

Inlet diameter, $d_1 = 1.2m$. $A_1 = \dfrac{\pi d^2}{4} = \dfrac{\pi \times 1.2^2}{4} = 1.131\,m^2$

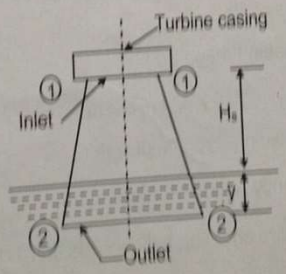
Output diameter, $d_2 = 180$ m, so $A_2 = \dfrac{\pi \times 1.8^2}{4} = 2.545\,m^2$

$V_2 = 3m/sec$

From continuity equation, $Q_1 = Q_2$

$A_1V_1 = A_2V_2$

$V_1 = \dfrac{Q}{A_1} = \dfrac{7.634}{1.131} = 6.75m/sec$



Head lost in draft tube $= 0.2 \times \frac{v_2^2}{2g} = \frac{0.2 \times 3^2}{2 \times 9.81} = 0.092\,m$

Applying Bernoulli's equation between section 1-1and 2-2

$$\frac{P_1}{\gamma} + (H_s + y) + \frac{V_1^2}{2g} = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + h_f$$

Now $\frac{P_2}{\gamma} = \frac{P_a}{\gamma} + y.$

So, $\frac{p_1}{\gamma} = \frac{P_a}{\gamma} - \left(\frac{V_1^2 - V_2^2}{2g}\right) + h_f - H_s$

$$= 10.30 - \left(\frac{6.75^2 - 3^2}{2 \times 9.81}\right) + 0.092 - 5.76 = 2.77m.$$

(ii) Efficiency of draft tube (h_d)

$$h_d = \frac{\frac{V_1^2 - V_2^2}{2g}}{\frac{V_1^2}{2g}} x100\%$$

$$= \frac{V_1^2 - V_2^2}{V_1^2} \times 100 = \frac{6.75^2 - 3^2}{6.75^2} \times 100 = 80.25\%$$

Solved Examples 17-2 :

Determine the overall efficiency of Kaplan turbine developing 2850 KW under a head of 5.20 m. It is provided with a draft tube with its inlet (diameter 3 m) set 1.8 m above the tailrace level. A Vaccum gauge connected to the draft tube indicates a reading of 5.20 m of water. Assume draft tube efficiency as 75%.

Efficiency of Kaplan turbine (η) =?

Output power (P) =2850 KW, H =5.2 m

Vacuum table reading = -5.2 m of water

Draft tube efficiency $(\eta_d) = 75\%$

Applying Bernoulli's equation between section 1-1 & 2-2

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + H_s + y = \left(\frac{P_a}{\gamma} + y\right) + \frac{V_2^2}{2g} + h_f$$

So, Assuming $P_a = 0$

$$\frac{P_1}{\gamma} = -\left(\frac{V_1^2 - V_2^2}{2g}\right) - H_s \Rightarrow \frac{P_1}{\gamma} = -\frac{\left(\frac{V_1^2 - V_2^2}{2g}\right)}{\frac{V_1^2}{2g}} x \frac{V_1^2}{2g} - H_s$$

Now, $\frac{\frac{V_1^2 - V_2^2}{2g}}{\frac{V_1^2}{2g}}$ = efficiency of draft tube $= \eta_d$.

$$\frac{P_1}{\gamma} = -\eta_d x \frac{V_1^2}{2g} - H_s$$

$$-5.2 = -0.75 x \frac{V_1^2}{2 \times 9.81} - 1.80.$$

$$\Rightarrow V_1 = 9.43 m / \sec.$$

$$Q = A_1 V_1 = \frac{\pi \times 3^2}{4} \times 9.43 = 66.66 m^3 / see.$$

$$\therefore \quad \text{Efficiency of turbine} = \frac{\text{Power Output}}{\gamma Q H} = \frac{2850}{9.81 \text{ x } 66.66 \text{ x } 5.20} = 83.80\%$$

Solved Examples 17.3

A proposed hydropower development having net head of 150 m design discharge of 25 m³/sec in going to use Francis turbine. Taking turbine efficiency 0.81, Calculate specific speed and turbine diameter and setting of the turbine.

Solution,

Net head (H) =150 m .

$Q_d = 25 m^3 / \sec, \ \eta = 0.81$

Specific speed (N_s) = ?

Turbine diameter (D) = ?

Power (P) $= \eta \gamma Q H$

$$= 0.81 \times 9.81 \times 25 \times 150 = 29.797.87 \, KW$$

$$= \frac{29.797.87}{0.746} = 39943.52 \text{ HP}$$

$$N_s = \frac{2400}{\sqrt{H}} = \frac{2400}{\sqrt{150}} = 195.96 rpm.$$

Rotational speed (N) $= N_s \frac{H^{5/4}}{\sqrt{P}}$

$$= \frac{195.96 \times 150^{5/4}}{\sqrt{39943.52}} = 514 \text{ r.p.m.}$$

Number of poles $P = \frac{120f}{N} = \frac{120 \times 50}{514} = 11.67 = 12.$

Corrected synchronous Speed, $N = \frac{120f}{p}$

$$= \frac{120 \times 50}{12} = 500 \text{ rpm.}$$

Corrected $N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{500 \times \sqrt{39943.52}}{(150)^{5/4}} = 190.36 \text{ rpm.}$

Diameter of turbine, $D = \frac{84.60\phi\sqrt{H}}{N}$

Where, $\phi = 0.0127\, N_s^{2/3} + 0.0275$

$$= 0.0127 \times (190.36)^{2/3} + 0.0275 = 0.0679$$

$$D = \frac{84.60 \times 0.679 \times \sqrt{150}}{500} = 1.407 = 1.50 \text{ m.}$$

Setting of turbine H_s

$$\sigma_c = 0.625\left(\frac{N_s}{38.78}\right)^2$$

Where, $N_s = \frac{N\sqrt{P}}{H^{5/4}}$ where, P in KW

$$= \frac{500 \times \sqrt{29727.87}}{(150)^{5/4}} = 164.42 \text{ rpm.} = 165 \text{ rpm}$$

$$\sigma_c = 0.625\left(\frac{165}{380.78}\right)^2 = 0.117$$

Again,

$$\sigma_c = \frac{H_a - H_v - H_s}{H}$$

$$-H_s = \sigma_c H - H_a + H_v$$

$$H_s = (H_a - H_v) - \sigma_c H$$

$$= 10 - 0.117 \times 150 = 7.60 m$$

Solved Examples 17-5

Select the size of Francis turbine for a site where the net head is 110 m and discharge is 140 m³/sec having efficiency if 94%. Determine also the elevation of turbine with reference to the water surface in tailrace. Assume the turbine will have to drive a 50 cycle generator.

Size of frames turbine (D) =?

Head, H=110 m

Design discharge, Q=140 m³/Sec

η =94%

Elevation of turbine with respect to tail water surface. Assume the turbine will have to drive 50 cycle Generator, f = 50Hz,

Setting of Turbine, H_s = ?

$P = \eta\gamma$ QH

$= 0.94 \times 9.81 \times 140 \times 110 = 142009.56$ KW

$= \frac{14009.56}{0.76} = 190261.34 HP$

Specific speed $N_s = \frac{2400}{\sqrt{H}}$ for Francis turbine

$= \frac{2400}{\sqrt{110}} = 228.831 = 229$ rpm

Synchronous speed $(N) = \frac{N_s H^{5/4}}{\sqrt{P}}$ Where P = Power in HP

$= \frac{299 \times (110)^{5/4}}{\sqrt{190361.34}} = 187$ rpm

No of pole $(N_p) = \frac{120 f}{N} = \frac{120 \times 50}{187} = 32.08 = 32$ Nos

Corredted $(N) = \frac{120 \times 50}{32} = 187.50$ rpm

Corrected $N_s = \frac{N\sqrt{P}}{H^{5/4}}$

$= \frac{187.80\sqrt{190361.34}}{(110)^{5/4}} = 229.64$ rpm $= 230$ rpm

Diameter of turbine $D = \frac{84.6\,\phi\sqrt{H}}{N}$ H

Where, $\phi = 0.0197 N_s^{2/3} + 0.0275$

$= 0.0197 \times (230)^{2/3} + 0.0275$

$= 0.767$

$D = \frac{84.6 \times 0.767 \times \sqrt{110}}{187.500} = 3.63 = 3.75m$

Setting of turbine

$$\sigma_c = 0.625\left(\frac{N_s}{380.78}\right)^2$$ For Francis turbine

Where N_s for P in KW

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{187.5\sqrt{142009.56}}{(110)^{5/4}} = 198.35$$

$$\sigma_c = 0.625\left(\frac{198.35}{380.78}\right)^2 = 0.17$$

Setting of turbine, $H_s = (H_a - H_v) - \sigma_c H = (10 - 0.17 \times 110) = -8.70$

Solved Examples 17-6

Design Pelton wheel turbine for a hydropower plant which have a net head 312.5m and discharge of 5 m^3/sec . Take the efficiency of the turbine is 85%, head H= 312.5, Design discharge, Q=5 m^3/sec, Overall efficiency, η =85 %

For Pelton turbine, velocity of jet $V_1 = C_d\sqrt{2gH}.$

$(C_v = 0.98)$

$= 0.98 \times \sqrt{2 \times 9.8 \times 312.50} = 76.74 m/\sec.$

$Q = A.V$

$$A = \frac{Q}{V} = \frac{5}{76.74} = 0.065 m^2$$

So, diameter of jet $d = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{0.065 \times 4}{\pi}} = 0.288m$

Diameter of runner (D) (Diameter of turbine)

$$take, \frac{D}{d} = 15 \Rightarrow D = 15 \times 0.288m = 4.32m$$

Number of bucket of runner $(Z) = 15 + \frac{D}{2d} = 15 + \frac{4.32}{2 \times 0.288} = 22.50 = 23 Nos$

Bucket spacing $= \frac{\pi \times 4.32}{23} = 0.59m = 59cm$

Design speed or Rotational speed (or speed of runner)

$N = \frac{60\,\overline{V}}{\pi D}$

Where, $\overline{V}$ = trangentail speed.

$\overline{V} = Kv.\sqrt{2gH} = 0.46\ \sqrt{2 \times 9.81 \times 312.50} = 36.02m/\sec$

So, $N = \frac{60 \times 36.02}{\pi \times 4.32} = 159.24$ rpm.

No of poles $N_p = \frac{120f}{N} = \frac{120 \times 50}{154.24} = 37.68 = 38$

So, Corrected $N = \frac{120f}{N_P} = \frac{120 \times 50}{38} = 158 rpm$

Power (P) = $\eta 9.81 QH = 0.85 \times 9.81 \times 5 \times 312.50$

$= 13028.90\,KW = \frac{13028.90}{0.746} HP = 17465.02\,HP$

Corrected $Ns = \frac{N\sqrt{P}}{H^{5/4}} = \frac{158\sqrt{17465.02}}{(312.50)^{5/4}} = 15.89 = 16$ rpm

Solved Examples 17-7

A hydroelectric plant built for design of H=20 m should develop power output 2000 HP. Determine the numbers of turbine units, if turbine having specific speed of N_s= 600 are to be used at a normal operating speed of N=230 rpm.

Solution,

H = 20 m.

Normal operating speed (N) = 230 rpm.

Specific speed (N_s) = 600 rpm

Power output (P) = 2000 HP.

No of turbine = ?

$$N_s = \frac{N\sqrt{P}}{H^{5/4}}, \quad \text{Where P in HP.}$$

$$\therefore\ P = \left(\frac{N_s\, H^{5/4}}{N}\right)^2 = \left[\frac{600 \times 20^{5/4}}{230}\right]^2 = 12173.68\,HP$$

Number of turbine required $= \frac{3000\,HP}{12173.68\,HP} = 0.16 = 1$

Solved Examples 17-8

Select turbine type and design with a power output 10 MW, head is 60 m and load variation up to minimum of 40%.

Here,

$P = 10MW = 10000KW = 13404.83HP.$

Load variation upto a minimum of 40%

Type of turbine =?

Corrected of turbine =?

For 60 m head both Kaplan and Francis turbine can be used. But due to the load variation upto a minimum of 40 %, Kaplan turbine must be selected because of the facility of the movable blades. The efficiency of the Kaplan turbine remains high over large range of load from 50% under load to 50% over load.

$$N_s = \frac{1475}{H^{1/3}} = \frac{1475}{60^{1/3}} = 378 \text{ rpm}$$

$$N = \frac{N_s H^{5/4}}{\sqrt{P}} = \frac{378 \times 60^{5/4}}{\sqrt{13464.83}} = 545 \text{ rpm}$$

$$\text{No of poles } N_P = \frac{120f}{N} = \frac{120 \times 50}{545} = 11.01 = 12 Nos$$

$$\text{Corrected } N = \frac{120 \times 50}{12} = 500 \text{ rpm}$$

$$\text{Corrected } N_s = \frac{500\sqrt{13464.83}}{60^{5/4}} = 347 \text{ rpm}$$

Solved Examples 17-9

For a hydropower project with installed capacity 20MW, head 500m and specific speed, N_s = 25, Design a turbine (diameter of nozzle, diameter of turbine).

Diameter of Nozzle (d) = ?

Diameter of turbine (D) = ?

Here, $N = \frac{60\overline{V}}{\pi d}$

For head, H=500 m, pelton turbine is used.

Where, $\overline{V} = K_u\sqrt{2gh} = 0.46\sqrt{2 \times 9.81 \times 500}$

$$So\ N = \frac{870.15}{d}$$

$$N_s = \frac{N\sqrt{P}}{H^{5/4}}$$

Where, P in HP

$$\text{so, } P = \frac{20000}{0.746} = 26809.65\ HP$$

$$\text{So, } \quad 25 = \frac{870.15\sqrt{26809.65}}{d\ (500)^{5/4}}$$

$$d = 2.41m$$

Taking, $\frac{D}{d} = 10$

$D = 10d = 2.41 \times 10 = 24.10m.$

* Determine the power of a pump lifting pulp with density of 1200 kg/m^3, discharge of 1100 l/sec, head of 60 m and efficiency of 70 %.

Soln:

$\gamma = \frac{9.81 \times 1200}{1000} = 11.772$ kN/m^3

$Q = 1100$ l/sec $= 1.1$ m^3/s

$H = 60$ m

$\eta = 70\% = 0.70$

Neglecting losses,

hydraulic power $= \gamma Q H = 11.772 * 1.1 * 60 = 776.952$

∴ the power of pump lifting pulp $= \frac{\text{hydraulic power}}{\eta}$

$= \frac{776.952}{0.70} = 1109.93$ KW //.